POWER

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Fixed Prices

IN the good old days you bought your suit of clothes for the lowest price you could induce the merchant to accept. If you were shrewder than he you came out ahead. It was the universal custom to wrangle and haggle over practically everything you bought.

One day a retail merchant came along who had the courage of his convictions. He set a definite value on every suit he had for sale and marked the price in plain figures. If you found a suit you liked and the price fitted your pocketbook, you bought. If not, there was the end of it.

People liked his plan; it saved time and it bred confidence. This man was successful and it was not long before all his competitors were following his lead. Today, the fixed-price plan is practically universal in many lines of trade. Automobiles are sold, not for what they will bring, but for a definite, predetermined price based upon the cost of manufacture, the quantity manufactured and the cost of selling.

The fixed price is not extensively used in the sale of many pieces of power-plant apparatus, although it could be used to advantage both for the manufacturer and the purchaser. Engines, turbines, dynamos, motors, boilers, pumps, stokers, and other similarly well standardized pieces of apparatus, all could be sold for a fixed price. There are few good reasons indeed why the manufacturers of such apparatus should not publish a set of net prices just the same as the clothing man, the furniture man, et al.

We know of dozens of cases where the engine man, the boiler man or the stoker man, as the case might be, has deliberately added 20 or 25 per cent. to the price he hoped to get, simply for the purpose of cutting it off again at the vigorous urging of the purchaser, thus permitting the latter to feel that he had done a fine stroke of business and saved himself some money. What a piece of monkey-shine for male adults to be indulging in!

The fault for such a state of affairs lies with both the manufacturer and the purchaser, although neither likes it over much. The former is reluctant to put in his best bid first for fear that if he does so, and by some lucky chance he has not the usual brisk competition, he may have to take less for his product than he could have obtained. The latter is so accustomed to find that he can play one bid against another to his own apparent advantage that he has never stopped to reason out the true state of affairs.

We predict that some manufacturer, growing tired of the present barbarous selling methods, will come out with a fixed-price policy. And we further predict that he will be so successful that others will rapidly fall in line.

Limiting Efficiencies of Steam Plants

By Prof. R. C. H. Heck

The influence of feed-water temperature upon absolute efficiency and calculations showing that the best efficiency attainable is that based on the steam consumption of the main engine with feed water at the temperature of its exhaust.

The low thermal expenditure of operating steam pumps is shown when their exhaust can be used to heat the feed water.

In every heat-engine plant there are two efficiencies to be considered—one absolute, the other relative. Absolute efficiency is the ratio of work output to heat input. It has two values in every case, first, the actual or realized value; second, the ideal. The latter represents the limit of the attainable (never reached), or the performance of an ideal heat engine under the controlling conditions. Dividing the actual by the ideal performance, gives the relative efficiency, sometimes called the efficiency ratio. These general ideas may well be illustrated by an example.

Example 1. Suppose that a condensing plant consumes 12.5 pounds of dry saturated steam per horsepower-hour at 165 pounds absolute pressure corresponding to 366.1 degrees Fahrenheit, from feed water at 149.1 degrees. From the steam tables, the total heat of one pound of dry steam at 165 pounds absolute is found to be 1195.4 B.t.u., while the heat in one pound of water at 149.1 degrees is 116.9 B.t.u.—both these quantities being measured above 32 degrees. Then the heat of formation, or that received from the fire per pound of steam is

$$1195.4 - 116.9 = 1078.5 \ B.t.u.$$

As to output, one horsepower for one hour equals

$$60 \times 33{,}000 = 1{,}980{,}000 \ foot\text{-}pounds$$

of work done. Reducing this to heat units by dividing by 778, the amount of heat converted into work is 2545 B.t.u. This work being credited to 12.5 pounds of steam, the output per pound is

$$2545 \div 12.5 = 203.6 \ B.t.u.$$

Then the absolute thermodynamic efficiency of the plant is

$$E = \frac{203.6}{1078.5} = 0.1888$$

or 18.88 per cent.

To obtain the ideal efficiency, the pressure or temperature of the exhaust steam must be known. Let this temperature be 90 degrees, corresponding to a pressure of 0.696 pound absolute, or to a vacuum of 28.50 inches when the barometer is at 29.92 inches. With heat received from the fire after the general manner of the ordinary boiler, or with the Rankine cycle of ideal steam-engine operation, the output would be 334.6 B.t.u. per pound of steam. (This quantity is the difference between the total heat in one pound of the steam at 165 pounds absolute and the heat after adiabatic expansion to 0.696 pound absolute.) If the ideal output, or the best that could be done under perfect conditions of working, is 334.6 B.t.u. per pound of steam, while the actual output is 203.6 B.t.u., the relative efficiency will be

$$E_r = \frac{203.6}{334.6} = 0.608$$

or 60.8 per cent.

It will be noted that only that part of the plant is considered which has to do with the use and application of heat, after it has been received from the fire. Boiler and furnace efficiency, in generating heat by combustion and in transferring it to the working medium (water and steam), is another matter.

Limit of Absolute Efficiency

The present purpose is not to show how ideal performance and efficiency are derived by logical processes, but to keep much nearer the practical side of the problem and consider how actual efficiency is related to feed-water temperature, and how the latter is determined by the combined influences of the main engine and of auxiliaries. Assuming the steam consumption to be fixed by operating conditions which are outside of this discussion, the following proposition is to be established:

If the heat of steam formation be determined from feed water of the temperature of the main exhaust, and if the steam consumption of the main engine alone be used in calculating the output, the resulting thermodynamic efficiency is the highest of which the plant is capable.

The best that can be done with any combination of auxiliaries and feed-water heater is to approach this upper limit. As stated, the proposition meets the simpler case in which the main engine can return heat to the boiler only from its exhaust; a broader specification, covering such action as the return of hot water from steam jackets, will be dealt with later.

Plant with Steam-driven Pumps

Example 2. Continuing the controlling data of the first example, let the 12.5 pounds of steam per horsepower-hour be divided into 11.8 pounds consumed by the main engine and 0.7 pound by the feed and condenser pumps. Then the limiting efficiency as just defined would be calculated for a steam consumption of 11.8 pounds, from water at 90 degrees. The heat input per pound of steam is the total heat minus the heat in the feed water, or

$$1195.4 - 57.9 = 1137.5 \ B.t.u.$$

while the output is

$$2545 \div 11.8 = 215.7 \ B.t.u.$$

then the best efficiency is

$$E_{max} = \frac{215.7}{1137.5} = 0.1896$$

or 18.96 per cent.

Suppose, first, that the exhaust from the pumps is turned into the main condenser or otherwise wasted, and that water is fed to the boilers at a hotwell temperature of 90 degrees. Under this condition the whole 12.5 pounds of steam must be charged, cutting the output to 203.6 B.t.u., while the input remains 1137.5 B.t.u. The resulting efficiency then is only

$$E = \frac{203.6}{1137.5} = 0.1790$$

or 17.90 per cent.

Of course, no properly arranged plant would operate in such a fashion; instead, the exhaust from the auxiliaries would be used to heat the feed water. Suppose that these pumps consume 45 pounds of steam per horsepower-hour of their own development, or convert into work

$$2545 \div 45 = 56.5 \ B.t.u.$$

per pound of steam. Assume further that 25 B.t.u. are lost by radiation from each pound of steam passing through the pumps. Then the total heat carried by the pound of pump exhaust will be the original heat, 1195.4 B.t.u., less 56.5 turned into work and 25 lost, or

$$1195.4 - 81.5 = 1114 \ B.t.u.$$

Temporarily disregarding the need of removing cylinder oil, let 0.7 pound of this exhaust combine with 11.8 pounds of hotwell water at 90 degrees; the resulting water will have some temperature t, to be found by equating the heat from the steam with the heat absorbed by the water, or by the equation

$$0.7\,[1114 - (t - 32)] = 11.8\,(t - 90)$$

whence

$$12.5\,t = 1864, \text{ or } t = 149.1 \ degrees$$

This is the feed temperature which was used in the first example.

An instructive comparison can be made by considering the heat input per horsepower-hour under different conditions. Thus for the main engine alone, which uses 11.8 pounds of steam and is capable of returning feed water at 90 degrees, or

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which requires 1137.5 B.t.u. per pound of steam, the heat received is

$$1137.5 \times 11.8 = 13{,}420 \ B.t.u.$$

For the combination of engine, pumps and feed-water heater, which uses 12.5 pounds of steam from water at 149.1 degrees, the corresponding heat supply is

$$1078.5 + 12.5 = 13{,}480 \ B.t.u.$$

From these numbers can be obtained the same efficiencies that have already been worked out with heat quantities per pound of steam, namely;

$$E_{max} = \frac{2545}{13{,}420} = 0.1896$$

and

$$E = \frac{2545}{13{,}480} = 0.1888$$

as in the first example.

The difference of 60 B.t.u. between the numbers 13,480 and 13,420 represents the thermal cost of operating the auxiliaries. It is about 0.7 of the 81.5 B.t.u. taken as the combined useful conversion and radiation loss of the steam. Comparing it with the supply 13,420, the percentage cost of the auxiliaries is only

$$60 \div 13{,}480 = 0.0045$$

or 0.45 per cent., as against an apparent steam cost of

$$0.7 \div 12.5 = 0.056$$

or 5.6 per cent.

In calculating these results, several minor sources of loss have not been taken into account. First of all, water from the condenser at full exhaust temperature—whether condensate or new water passed through a surface heater ahead of the condenser—is hardly attainable; there will probably be a deficiency of at least five degrees, and it may be much larger. An oil separator will remove from the pump exhaust considerable water, which will carry some heat with it. Radiation from the feed pump and feed piping will also cause a small loss.

Power-driven Pumps

The scheme which has been under discussion may now be compared with that of using power-driven pumps. These are either coupled to the main engine, as in many pumping-engine and marine plants, or are driven by motors as in some steam-electric plants. Whatever power they absorb is either directly or indirectly subtracted from the useful output of the unit. So far as the engine side of the plant is concerned, elimination of steam-driven pumps means the return of feed water at or below exhaust temperature. In the second example the first case, that of 11.8 pounds of steam per horsepower-hour, would correspond to the feed water at a maximum temperature of 90 degrees; but, as just stated, the efficiency of 0.1896 or a little less at the engine would be diminished to the extent of the power consumed by the pumps. The use of a live-steam heater or of an injector, either one giving the boiler the advantage of hot feed water, would not appreciably affect the thermodynamic relations, since both simply take heat from the boiler and put it back again. If an economizer be included in the plant, it would be considered a part of the boiler, sharing in the function of abstracting heat from the fire.

The contrasting points of the two types of arrangement may be summarized as follows:

With steam pumps, provided that they are not excessively wasteful, all their exhaust may be condensed by the main feed, and the heat rejected by the pumps may thus be returned into the main circuit. Consequently, the power developed and applied by these auxiliaries costs only its own thermal equivalent, plus secondary losses in oil removal and in radiation. If the latter could be reduced to zero, these minor members of the plant would have unit efficiency.

With power pumps, the percentage of power consumed will be much less than the percentage of steam used by the other type of auxiliaries; but since this power is produced at only the efficiency of the main engine, say from 17 to 22 per cent. in good practice, its thermal cost is much higher than in the other case.

The conditions assumed in the example correspond with a pump power equal to about 1.6 per cent. of that of the engine, as may be shown by the following example:

With independent steam-driven pumps, the pump output per engine horsepower requires 0.7 pound of steam, at the hourly rate of 45 pounds per pump horsepower. Using the power developed at the engine rate of about 12 pounds per hour, the work of the pumps would be done by

$$\frac{12}{45} \times 0.7 = 0.19 \ pound$$

of steam, which is 1.6 per cent. of 12 pounds.

One and six-tenths per cent. of 2545 B.t.u. is 40 B.t.u.; and with a thermodynamic efficiency of 0.19 for the engine, this would require a heat supply of more than 200 B.t.u. This is high compared with the 60 B.t.u. deduced in the second example for steam-driven pumps with complete utilization of the pump exhaust.

Feed Temperature

In short, what has been discussed is the influence of feed-water temperature upon thermodynamic efficiency. The point emphasized is the determinative influence of the main engine, or of the temperature which it alone (with its condenser) can give to the water delivered to the boiler. In the simpler cases the limiting or ideal feed temperature from the engine (never quite attained) is that of the exhaust steam. If the engine has steam jackets or reheaters in its receivers, and can therefore return hot jacket water, the ideal feed temperature is higher.

Example 3. With the engine using 11.8 pounds of steam per horsepower-hour, let 10 per cent. be condensed in the jackets and reheater, for simplicity at the full initial temperature of 366.1 degrees, the exhaust temperature remaining 90 degrees. What will be the ideal feed-water temperature due to the engine?

Solution. At 366.1 degrees the heat in one pound of water is 338.1 B.t.u. above 32 degrees, or

$$338.1 - 57.9 = 280.2 \ B.t.u.$$

above water at 90 degrees. The 1.18 pounds of jacket water, if there were no lowering of its temperature by radiation, would carry 330.6 B.t.u. above 90 degrees. This would raise the whole 11.8 pounds of feed water (including that from the jackets) through

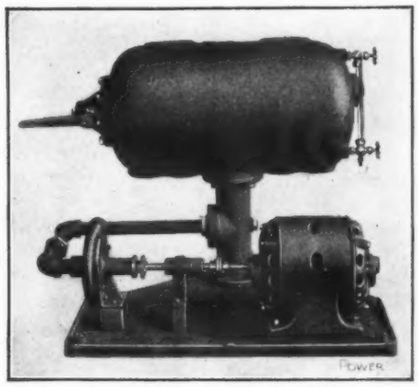
$$330.6 \div 11.8 = 28.1 \ degrees$$

or to 118.1 degrees Fahrenheit, which is therefore the ideal feed temperature sought. Diminishing the input by 28.1 B.t.u., or from 1137.5 to 1109.4 per pound of steam, while keeping the output at 215.7, the limiting efficiency is raised to

$$E_{max} = \frac{215.7}{1109.4} = 0.1945$$

Motor Driven Pump and Receiver

On the motor-driven centrifugal pump and receiver shown herewith the lever arm at the end of the cast-iron receiver is balanced on the inside of the receiver by a float and the position of the arm is changed with the rise or fall of the water



Pump and Receiver

level. The outside end of the arm is attached to the electrical device controlling the motor. When the water has risen in the receiver to a predetermined point the motor is started automatically and the water is pumped out. The set is manufactured by the Advance Pump & Compressor Co., Battle Creek, Mich.

Substituting Gas and Oil for Coal

By C. A. Tupper

The Fort Smith Light & Traction Co., which took over the properties of two competing local companies in Fort Smith, Ark., some years ago, offers an interesting example of systematic improvement in power-plant operation, particularly as the results obtained were secured without heavy additional investment. Among these changes the most interesting to the practical operating man is the use of natural gas under the boilers, displacing coal, and the installation of an auxiliary oil-burning system.

Furnaces

Each of the furnaces as now equipped has 15 burners of the Kirkwood type, consuming about 25 cu.ft. of gas per second when turned on full. The supply is brought to the furnace by 8-in. piping, from which 2-in. tubes branch off to the burners. Here the required heat is regulated to a certain extent by the length of the flame, but principally by the number of burners in service. Air mixture is effected by a rotating chamber, forming part of the burner, through which the gas passes, and by means of a suitable gate opening in the mixing chamber, the admission of air is regulated so that the gas burns with a bright blue flame. Provision has also been made for automatic regulation of the gas supply from the boilers.

Between and underneath each five gas burners is an oil spray nozzle. This system, taking about ¼ gal. of oil per horsepower, is utilized whenever the demand for the gas, which is largely consumed locally, causes the pressure to fall off, or anything else occurs tending to interrupt the supply. The oil burners are also of the Kirkwood type, manufactured by Tate, Jones & Co., of Pittsburg. Live steam from the boilers, reduced through a valve to 35 lb. pressure, is mixed with the oil in an annular chamber and from the latter the jet emerges through a flattened nozzle, giving a "fish-tail" effect to the flame.

Boilers

The five boilers are of the Aultman & Taylor type, the furnaces being fitted with brick checkerwork. In the boiler room only one head fireman and an attendant are now required for a 12-hour shift, whereas four or five times that number would be necessary if coal were used as fuel. Boiler-feed water is taken from the city mains as any supplied from the Arkansas River would have to be filtered, and this also dispenses with feed pumps. In the steam system a pressure of 125 lb. is maintained.

Interior of Boiler Room, Showing Gas and Oil Burners

Engine Room

In the engine room are installed noncondensing engine-driven units aggregating 5200 hp. These are divided between two-phase, 60-cycle, 2300-volt generators for commercial power and lighting service, and direct-current railway generators wound for 550 volts. There is also a 350-kw. motor-generator set, through which current for either system can be taken from the other.

The great gain effected by the introduction of gas and oil in place of coal has been not so much in the lower cost of the fuel as in the simplicity and convenience of operation, with greatly decreased attendance. There have also been eliminated the expense of handling coal and ashes, of which the cost of bringing the gas and oil to the furnaces is but a fraction. Furthermore, deterioration has been reduced to a minimum and there is a total absence of dust and smoke.

Oil System

The oil used in the plant is kept in a ventilated underground tank, surrounded by coils through which exhaust steam can be passed in winter when necessary to maintain the proper temperature. Duplex pumps are used for forcing the oil to the furnaces, with suitable regulation of pressure; and relief valves in the piping system, set at 45 lb., prevent an oversupply, with an automatic return to the tank, including all drips or overflow. The natural gas comes in directly from the field pipe line and enters through a reducing valve at 15 lb. pressure. Both the oil and gas are metered in transit.

In supplying and controlling the fuel care has been taken to avoid danger from fire or explosions, including a steam blanketing system and arrangements for quickly emptying the oil tanks in case of hazard.

Boiler Explosions at Sea

The danger of boiler explosion on board ship is not as serious as the same danger ashore, in power houses or factories and office buildings, says Capt. Hobson in the *Engineering Magazine*.

Due precautions, particularly in the efficiency of fire-room personnel, will reduce this danger to a minimum of only secondary importance, in most cases. But this cannot be said of the "Titanic"; her boilers were out close to the sides of the ship, with no inner skin to protect them. The boiler spaces extended for over a third of the ship and the bodies of the boilers themselves offered a target covering a considerable part of this space. A craft of comparatively small size, colliding perpendicularly or even obliquely in this space, would have easily reached a boiler.

In collision, the "Titanic" ran about as much danger from being blown up by a boiler explosion as from sinking from the inflow of water. No boiler located in the wings should be left without the protection of a partial longitudinal bulkhead, or at least an inner skin on the side. This risk on the "Titanic," though not great, was unnecessary. At very little cost and no difficulty, the inner skin could have been carried up the sides in the boiler spaces as far as the lower deck. In fact, it would have been good protection to take the same precaution in the engine-room space.

Combustion and Flue Gas Analysis

By Charles M. Rogers

Brief outline showing theoretical relation between percentage of CO_2 and boiler efficiency.

Some charts giving actual results of flue-gas analyses.

The predominating factor in the economical operation of a steam-generating station is combustion, and on the average, over 50 per cent. of the money spent for the production of power is consumed in supplying the boiler room. It would seem, then, that this is the end to "get after," if for no other reason than a monetary one. A prominent power superintendent told the writer not long ago that if he built any more power plants he would put the engineer's office in the boiler room.

The greatest loss in the operation of a steam boiler is made up of two factors, viz.: (1) the sensible heat of the escaping flue gases, and (2) the potential heat of the combustible constituents of these gases. This loss varies from 10 to 50 per cent. of the total heat contained in the fuel, and as a large percentage of this loss is avoidable, it would seem that too much study and attention could not be given to the subject of economical combustion.

Loss Due to Excess Air

It is readily seen that by increasing the air supply above the amount theoretically required for perfect combustion, the number of heat units carried up the stack by the waste gases also increases. The number of units lost in this way in the hot gases is determined by multiplying the difference between the temperatures of the stack gases and the air supplied to the furnace by the number of pounds of air supplied per pound of coal, plus the weight of combustible burned per pound of coal, times the specific heat of the gases. The loss varies, then, directly as the stack temperature. The excess air reduces the percentage of CO_2 in the gases, and makes it equal to the maximum possible percentage of CO_2, divided by the number of times the theoretically required amount of air is supplied, when no CO is present. This loss varies therefore inversely as the percentage of CO_2 in the gases.

The loss caused by the gases carrying off potential heat contained in the combustible constituents would naturally not depend upon the temperature of the gases. And the stack temperature alone can hardly be relied upon as indicative of high or low efficiency in that so many factors, such as dirty heating surfaces, air infiltration, etc., enter as functions of its rise and fall.

Since the sensible heat lost in the waste gases decreases as the percentage of CO_2 increases, then, other things being equal, a high percentage of CO_2 is conducive to high efficiency.

The curves in Fig. 1 were plotted from actual results obtained with a boiler equipped with a water meter, coal weigher and CO_2 recorder. They serve as a good illustration of how the boiler efficiency varies directly as the percentage of CO_2, and are but one case of many that show this relation distinctly.

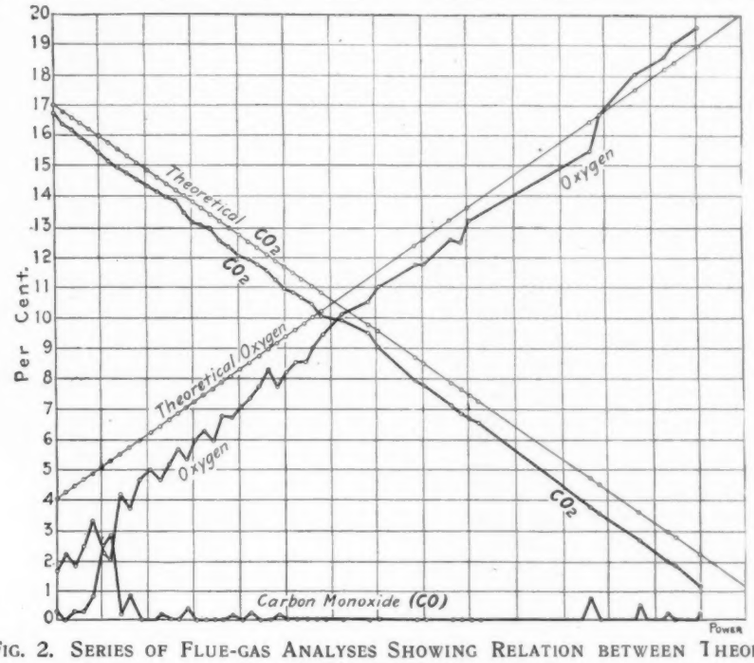


Fig. 2. Series of Flue-gas Analyses Showing Relation between Theoretical and Actual Percentages

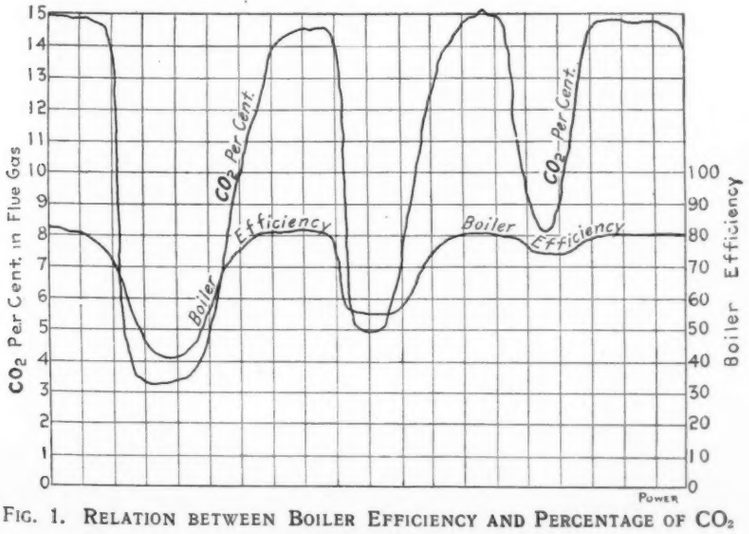


Fig. 1. Relation between Boiler Efficiency and Percentage of CO_2 in Flue Gases

Incomplete Combustion

It is well known that with a high percentage of CO_2 there is danger of an accompanying loss from incomplete combustion due to the formation of CO. This depends to a great extent upon the furnaces, fuel, draft and other general conditions. Apropos of this, the proper thing to do is to attain a comparatively high percentage of CO_2, which represents a gain that more than offsets any loss due to incomplete combustion.

Unfortunately, the writer has not been able to find any definite relation existing between the CO_2 or O and CO. The sum of these three constituents varies with the completeness of combustion, composition of the fuel, and to some ex-

tent with the temperature at which the gases are analyzed, as moisture is calculated as nitrogen and therefore brings the sum below 21. Since all the oxygen which combines with hydrogen forms water vapor, which is condensed, for a fuel which contains hydrogen, as does coal, the sum of the oxygen components will always be less than 21 per cent.

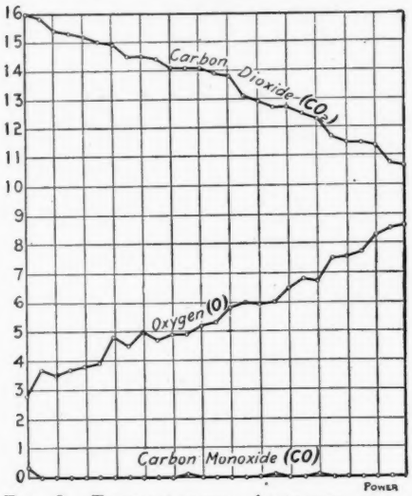


FIG. 3. TWENTY-SEVEN ANALYSES FROM 500-HP. BOILER

Thus, it will be noticed upon referring to Figs. 2, 3, 4 and 5, that by adding together the percentages of the oxygen components at the left of the chart, the sum rarely exceeds 19, while on the right in Fig. 2 it approaches 21, the percentage of oxygen in the air. This is what would be expected, as the readings on the right were taken while the fires were banked; the moisture, hydrogen and other constituents, which unite with part of the free oxygen of the air, having been expelled, and therefore leaving the total percentage of oxygen available for combining with the carbon to form CO_2 and CO. This checks up surprisingly close with the results of other tests not shown here.

Figs. 2 and 4 indicate that there is more danger of the formation of CO with the percentage of CO_2 above 13. These analyses were made with an improved type of Orsat hand analyzer. It is of interest to note that in Fig. 3, which gives the results obtained with furnaces in another plant but of the same make where the conditions are about the same, with the exception that the dampers and stoker engines are controlled automatically by the steam pressure, a trace of CO is found only four times out of 27 trials, even when the CO_2 ran as high as 15 and 16 per cent.

The results plotted in Fig. 4 were obtained from boilers where live steam was admitted through 6 jets distributed along the front of the furnace just above the coking grates. The results plotted in Fig. 5 are from the same boilers with the jets shut off. A study of these two charts shows that the possibility of CO formation is greater with the jets on when the CO_2 approaches the maximum; but while getting the normal percentage of CO_2, which averages about 12 per cent. with these boilers, there is little danger of the jets causing the formation of CO.

At the time each sample was taken, from which the results plotted in Fig. 3 were obtained, the conditions of the fire, draft and smoke were noted, and a comparison of these data reveals the fact that the CO (on the left-hand side) was found, during the emission of smoke only, when the fires were too heavy, or when the draft had been suddenly checked or green coal had been admitted in excess.

The straight lines in Fig. 2 give the theoretical percentages of CO_2 and O. That is, for any given percentage of CO_2 the theoretical O will be found on the

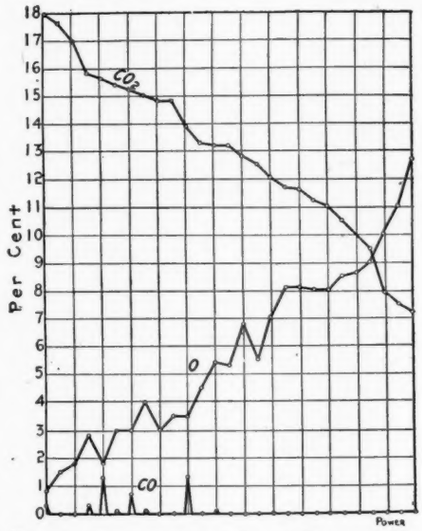


FIG. 4. ANALYSES OF FLUE GASES FROM BOILER USING STEAM JETS

same ordinate, and their sum will be equal to the percentage of O in the air.

LOSS DUE TO CO

It has been calculated that the average theoretical loss approximates 4.8 per cent. for every 1 per cent. of CO found in the flue gases. This and the following will illustrate the importance of guarding against the formation of CO.

The loss due to the formation of CO in reality amounts to considerably more than the foregoing theoretical figure indicates; as hydrogen and hydrocarbons usually escape unconsumed under the same conditions as does CO. On account of the analytical difficulty in determining the small quantities of these high heat-value combustibles in the flue gas, no practical examples are given which represent the exact losses due to their escape. The government, however, has found by several series of tests that the efficiency of a boiler drops 20 per cent. for every 1 per cent. of CO found in the waste gases. About three-quarters of this 20 per cent. loss is accounted for by the escape of combustibles other than CO. This figure is sufficiently high to be questioned by some, but it seems to be the case nevertheless, and furnishes an incentive to secure furnace operation free from the production of CO.

CONCLUSION

The foregoing tends to prove, then, that (1) the percentage of CO_2 in the escaping gases may be relied upon as an index of (a) high or low combined efficiency, and (b) the approximate percentage of heat carried off by the gases. (2) A high or low stack temperature is not always indicative of low or high efficiency. (3) A large part of the sensible and potential heat lost, ranging from 10 to 50 per cent. of the total heat contained in the coal, and in some cases even more, is avoidable. (4) No easily fixed relation exists between the percentages of CO_2 or O and CO, but knowing the percentage of CO_2, that of the O is easily determined approximately, or *vice versa*. (5) The occurrence of CO in the waste gases is a decided danger signal; not so much on account of the loss which it represents, but because it is usually accompanied by hydrocarbons and hydrogen, which are high in heat value, ranging from 21,000 to 62,000 B.t.u. per

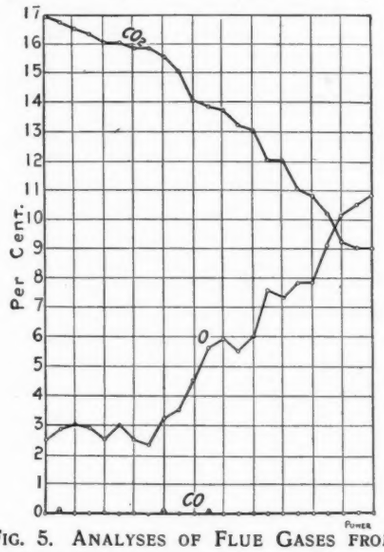


FIG. 5. ANALYSES OF FLUE GASES FROM SAME BOILER AS IN FIG. 4 WITHOUT STEAM JETS

pound. (6) Certain equipments and conditions, other things being equal, permit the regulating of a furnace so as to produce a comparatively high CO_2 without the formation of CO while others work adversely. (7) With a normal percentage of CO_2, the CO caused by steam jets is negligible. (8) When dense smoke is emitting from a stack, the loss is not indicated by the small percentage of precipitated carbon which makes the smoke visible, but by the loss of the unconsumed combustibles which usually accompany such conditions; a smokeless stack may be carrying away large amounts of high heat-value combustibles.

Heating Surface, Boiler Horsepower

By F. L. Johnson

Brief directions for computing the heating surface and rated horsepower of horizontal return-tubular boilers and instructions for using the tables which have been prepared for the purpose of materially shortening the operation.

At a recent engineers' examination in Ohio, this question was asked one of the candidates for a license: "What is the heating surface and horsepower of a horizontal return-tubular boiler 60 in. in diameter, 16 ft. long, having forty-four 4-in. tubes?" The heating surface of a boiler is that portion which is exposed to the heat of the furnace and to the hot products of combustion on one side and is covered by water on the other.

In the horizontal return-tubular boiler the water-covered surface exposed to heat is made up of the lower half of the shell, the inside area of all of the tubes and the area of one head less the area taken up by the tubes in both heads.

In these calculations one-half the area of the shell is taken because that part of the shell below the line where the brickwork is closed in and comes in contact with the shell and limits the height to which the fire or hot gases may rise closely approximates one-half of the circumference of the shell.

In calculating the heating surface of the tubes, the interior or fire surface is taken because the water will take away from the outside and greater area all of the heat that can be transmitted through the tube from the inside and smaller area.

The area of only one head is reckoned because the total surface of both heads which are exposed to heat is but little, if any, more than this. At the rear the setting is closed in just above the upper row of tubes, cutting off nearly one-half the area of the head, and at the front the surfaces below the tubes and above the water line transmit no heat to the water. This is practically equal to the area of one head. From this area must be subtracted the cross-sectional area of the tube ends in both heads as they present no surface whatever to absorb the heat of the passing gases.

Horsepower is a rate of work and in steam engines is measured by the foot-pounds of energy developed in a given time. In a steam boiler the energy developed cannot be measured or expressed in foot-pounds; for the work done is the conversion of water into steam and must be measured in terms of heat units.

In 1876, the committee of judges of the Centennial Exposition, in order to correctly compare the performance of many competing boilers, formulated what is known as the Centennial rule for the horsepower unit of steam boilers. This unit is the evaporation of 30 lb. of water per hour into steam at 70 lb. gage pressure from feed water at a temperature of 100 deg. F., an equivalent of the transfer of 33,305 heat units from the furnace to the water. This rule was accepted in 1884 by a committee of the American Society of Mechanical Engineers, which recommended its adoption by the society as a standard for use in all boiler trials.

As the value of the heating surface of a boiler depends on the quantity of heat generated in the furnace per square foot of heating surface of the boiler, the number of square feet required to develop a boiler horsepower will depend on the conditions of operation in each case.

From the earlier days of steam-boiler practice until comparatively recent times, the rates of combustion were moderate, and 15 sq.ft. of heating surface was the common allowance made for one boiler horsepower. By common consent, it was dropped to 12 sq.ft. for horizontal return-tubular boilers and 10 sq.ft. for water-tube boilers.

Some years ago, the New England Boiler Builders Association adopted 10 sq.ft. as the area required to develop one horsepower in horizontal return-tubular boilers.

Taking the boiler given in the question, the total area of the shell is found by multiplying the diameter by 3.1416 and by its length; 60 in. is 5 ft., hence the surface of one-half the shell is

$$\frac{5 \times 3.1416 \times 16}{2} = 125{,}664 \; sq.ft.$$

The inside diameter of a standard 4-in. boiler tube is 3.74 in., which multiplied by 3.1416 gives the inside circumference

$$3.74 \times 3.1416 = 11.75 \; in.$$

Hence each foot of length of a 4-in. boiler tube will contain

$$\frac{11.75 \times 12}{144} = 0.979 \; sq.ft.$$

of heating surface, and in 44 tubes, each 16 ft. long, there will be

$$16 \times 44 \times 0.979 = 689.216 \; sq.ft.$$

The area of a 60-in. (5-ft.) boiler head is

$$5 \times 5 \times 0.7854 = 19.635 \; sq.ft.$$

The cross-sectional area of a 4-in. boiler tube is 12.566 sq.in., and in 88 tube ends there will be

$$\frac{88 \times 12.566}{144} = 7.679 \; sq.ft.$$

Therefore, the heating surface of the boiler will be, in the shell,

$$\frac{1}{2} \times 5 \times 3.1416 \times 16 = 125.664 \; sq.ft.$$

In the tubes

$$16 \times 44 \times 0.979 = 689.216 \; sq.ft.$$

In the heads

$$19.635 - 7.679 = 11.956 \; sq.ft.$$

adding

$$125.664 + 689.216 + 11.956 = 826.836 \; sq.ft.$$

If 10 sq.ft. of surface are allowed per horsepower, the boiler will be rated as

$$\frac{826{,}836}{10} = 82.68 \; hp.$$

TABLE 1. PRINCIPAL DIMENSIONS OF STANDARD BOILER TUBES FROM 2 TO 4 IN. OUTSIDE DIAMETER

External Diameter	Standard Thickness	Inside Diameter	Inside Circumference	Outside Circumference	Heating Surface per Foot of Length	Length per Square Foot of Heating Surface	Cross Section in Square Feet	
							Outside Diameter	Inside Diameter
2	0.095	1.810	5.6858	6.2832	0.4739	2.11	0.0218	0.017870
2¼	0.095	2.060	6.4717	6.7066	0.5393	1.86	0.0276	0.023112
2½	0.109	2.282	7.1691	7.8540	0.5974	1.67	0.0341	0.028402
2¾	0.109	2.532	7.9545	8.6394	0.6629	1.51	0.04120	0.034870
3	0.109	2.782	8.7399	9.4248	0.7283	1.37	0.04909	0.041451
3¼	0.120	3.010	9.4562	10.2102	0.7880	1.27	0.05761	0.049412
3½	0.120	3.260	10.2416	10.9950	0.8535	1.17	0.06681	0.05796
3¾	0.120	3.510	11.0270	11.7810	0.9189	1.08	0.0767	0.067195
4	0.134	3.732	11.7244	12.5660	0.9790	1.02	0.0872	0.075270

TABLE 2. AREAS TO BE DEDUCTED FROM AREA OF BOILER HEAD FOR ANY NUMBER OF TUBES FROM 2 TO 4 IN. DIAMETER

External Diameter of Tubes	1	2	3	4	5	6	7	8	9
2	0.0218	0.0436	0.0654	0.0872	0.1090	0.1308	0.1526	0.1744	0.1962
2¼	0.0276	0.0552	0.0828	0.1104	0.1380	0.1656	0.1932	0.2208	0.2484
2½	0.0311	0.0682	0.1023	0.1364	0.1705	0.2046	0.2387	0.2728	0.3069
2¾	0.0412	0.0824	0.1236	0.1648	0.2060	0.2472	0.2884	0.3296	0.3708
3	0.0491	0.0982	0.1473	0.1964	0.2455	0.2946	0.3437	0.3928	0.4419
3¼	0.0576	0.1152	0.1728	0.2304	0.2880	0.3456	0.4032	0.4608	0.5184
3½	0.0637	0.1274	0.2011	0.2548	0.3185	0.3822	0.4459	0.5096	0.5733
3¾	0.0767	0.1534	0.2301	0.3068	0.3835	0.4602	0.5369	0.6136	0.6903
4	0.0872	0.1744	0.2616	0.3488	0.4360	0.5232	0.6104	0.6976	0.7848

TABLE 3. HEATING SURFACE IN HALF SHELL OF CYLINDRICAL BOILERS

Length in Feet	Diameter of Shell, in Inches																					
	42	44	46	48	50	52	54	56	58	60	62	64	66	68	70	72	74	76	78	80	84
14	76.96	80.63	84.29	87.96	91.64	95.30	98.95	102.63	106.29	109.96	113.62	117.28	120.95	124.62	128.28	131.94	135.61	139.28	142.94	142.94	150.27
14.5	79.71	83.51	87.30	91.10	94.91	98.70	102.48	106.29	110.09	113.88	117.68	121.48	125.27	129.07	132.86	136.66	140.46	144.25	148.05	148.05	155.64
15	82.46	86.39	90.31	94.24	98.18	102.10	106.02	109.96	113.88	117.81	121.74	125.66	129.58	133.52	137.44	141.37	145.30	149.23	153.15	153.15	161.00
15.5	85.21	89.27	93.32	97.38	101.46	105.51	109.55	113.62	117.68	121.73	125.80	129.85	133.90	137.98	142.03	146.09	150.14	154.20	158.26	158.26	166.37
16	87.96	92.15	96.33	100.53	104.73	108.91	113.09	117.29	121.47	125.66	129.85	134.04	138.22	142.43	146.61	150.79	154.99	159.17	163.36	163.36	171.74
16.5	90.71	95.03	99.34	103.67	108.00	112.31	116.62	120.95	125.27	129.58	133.91	138.23	142.54	146.88	151.19	155.51	159.83	164.15	168.47	168.47	177.11
17	93.47	97.91	102.35	106.81	111.28	115.72	120.16	124.62	129.07	133.52	137.97	142.42	146.86	151.32	155.77	160.21	164.67	169.12	173.57	177.99	186.93
18	98.95	103.67	108.37	113.09	117.82	122.52	127.22	131.95	136.66	141.37	146.09	150.80	155.50	160.22	164.93	169.64	174.36	179.07	183.78	188.46	197.92
19	104.45	109.43	114.39	119.37	124.37	129.33	134.29	139.28	144.25	149.23	154.20	159.17	164.14	169.12	174.10	179.06	184.05	189.02	193.99	198.93	208.92
20	109.94	115.19	120.42	125.66	130.90	136.14	141.36	146.61	151.84	157.08	162.32	167.55	172.78	178.02	183.26	188.48	193.73	198.97	204.20	209.40	219.92
Area of head, sq.ft.	9.62	10.56	11.54	12.57	13.64	14.75	15.90	17.10	18.35	19.63	20.97	22.34	23.76	25.22	26.73	28.27	29.87	31.50	33.18	34.91	38.49

Tables 1, 2 and 3 have been prepared for the purpose of shortening the process of calculating the areas of heating surface. In Table 1 will be found all of the principal dimensions of standard boiler tubes from 2 to 4 in. in diameter. Table 3 gives the heating surface of one-half the shell and the area of the heads of boilers from 42 to 84 in. in diameter and from 14 to 20 ft. in length. Table 2 gives the areas to be deducted from the area of the boiler head for various sizes and numbers of tubes.

To illustrate the use of the tables, take the same example, Shell, 5x16 ft.; forty-four 4-in. tubes.

Opposite 16 and under 60 in Table 3, the number 125.66 is found. To find the total cross-sectional area of the eighty-eight 4-in. tube ends to be deducted from the area of one head, use Table 2. Under 8 and opposite 4, 0.6976 will be found. This is the area in square feet of eight 4-in. tube ends. By moving the decimal point one place to the right, the number is multiplied by 10 and becomes 6.976, which is the area of 80 tube ends, which added to the other number gives the total area desired.

$$0.6976 + 6.976 = 7.67 \textit{ sq.ft.}$$

Or the area of one tube end 0.0872 square may be multiplied by the number of tube ends.

$$88 \times 0.0872 = 7.6676$$

Adding the area as found in the tables, the total heating surface of a 5x16-ft. boiler having forty-four 4-in. tubes is found to be:

In one-half the shell, 125.66 sq.ft.
In forty-four 4-in. tubes, 689.21 sq.ft.
In the heads, 11.96 sq.ft.
making a total of

$$125.66 + 689.21 + 11.96 = 826.83 \textit{ sq.ft.}$$

Wrong Gage Connection

By Victor White

The following incident shows how needless delays are sometimes caused, occasioning great inconvenience, through a comparatively simple and easily avoidable mistake.

A certain power house was laid out and run on the plan of each engine having its own condensing plant, there being an alternative exhaust into the atmosphere through an automatic valve on each engine exhaust. An additional unit had just been installed and had passed through its preliminary slow- and full-speed runs on atmospheric exhaust in order to make sure that everything was working properly. Before putting the load on, it was decided to run the engine light with the condenser to see that this part of the equipment was in order. It should be explained that for the convenience of the attendant there was mounted on the wall at the back of each engine a 12-inch dial showing the vacuum; these were connected by means of copper tubes to each condenser in the basement.

Before throwing this particular set onto the condenser, a reading was taken on the vacuum gage on the wall, but, although circulating water was flowing freely through the condenser and everything to all appearances appeared to be in order, no vacuum was indicated. The engineers studied this problem for some time; the engine was shut down and the pipe leading from the condenser to the gage was taken apart and tested to see whether there was a leak or if it were choked anywhere, but everything seemed in good order.

The pressure gage was next taken off the wall and put directly on to the condenser itself, and when the engine was started and the air and circulating pumps attached to it were working, the gage immediately gave a reading of 28 inches. It was taken off again, put on the wall and the connecting pipe put back into position, when, as before, it registered nothing.

Three days were spent in trying to solve the mystery when finally a fitter's helper, hunting around underneath the engine, discovered that the plug and nipple attached to the connection between the condenser and the engine had been put in on the engine side of the condenser valve instead of next to the condenser. The result was, of course, that as the steam of the engine was passing to the atmosphere before the condenser valve was opened, and as the engineers had not ventured to open the condenser valve until they were sure everything was correct, the gage was simply registering the pressure of the exhaust steam to the atmosphere.

The gage pipe was disconnected, the hole in the exhaust pipe was plugged, a new hole was drilled and tapped next to the condenser, the pipe put into position, and then the readings on the gage were found to be satisfactory.

The Electric Light in 1878

In a report of the Bombay Gas Co., 34 years ago, says the *Scientific American*, a member who shortly before attended an exhibition of the electric light stated that this light was not one "which any man could put about his house and leave in the care of his servants. There could be no store of electricity, and the instant the engine ceased to work from any cause or accident all the lights would be gone."

Another attendant at the meeting "was quite ready to admit that at some distant day electric lighting might become general; but it might be that at some remote period men might fly. . . . All these things might be possible, but they were a long way off."

Testing an Engine under Difficulties

By H. B. Lange

A brief description of the methods followed to make sure that only the correct quantity of water should be charged to an engine which was being tested for steam consumption.

Often before an engine is paid for by the buyer a test is made to determine its efficiency and steam consumption. On such a test both the buyer and builder are represented. The builder in submitting his proposal guarantees that the engine will consume not more than a certain amount of steam per indicated horsepower or kilowatt-hour with a given steam pressure at the throttle and with a certain back pressure or vacuum. It is, therefore, necessary that an economy test be made to determine whether the engine meets the guarantee, and it is essential that the conditions under which the test is run be satisfactory to both parties involved and that the data and measurements be accurate within reasonable limits.

It is generally thought that it is an easy matter to make such an economy test; that all that is necessary is to weigh or meter the feed water, indicate the engine and read the switchboard instruments, and possibly to account for the steam used by the feed pump in some simple way and make corrections in the feed-water calculations. The testing engineer, however, knows from experience that this is by no means the case; that the first requirement is that his measuring instruments, such as gages, water meters, scales, thermometers, indicator springs, electrical meters, etc., must be correct, or if they are not he must know to what extent they are out. Few, if any, instruments are 100 per cent. correct and therefore they must be proved and calibrated in connection with standard instruments which are known to be correct. This calibration of instruments, while a more or less difficult task, is the least of the engineer's troubles when it comes to making a commercial test. The special and local conditions connected with each test which give the greatest trouble and make the most work are never the same, yet they occur in as many different cases as there are plants or engines to be tested. The most serious difficulties that come up are due to the condition of the piping systems; they are generally well laid out and erected accordingly, but the layouts are usually made for the best operating conditions and do not provide in any way for testing, and when it comes to setting up the various measuring instruments for the test the greatest difficulties are met in this part of the plant. The problems that confront the engineer in this connection are in some cases difficult to solve, yet they must be met and overcome in some way. This is illustrated by a test in connection with the plant in a large office building which consisted essentially of three direct-connected units, three water-tube boilers and two direct-acting duplex feed pumps. The boilers and engines were placed in the relative positions shown in the diagram. Engines Nos. 1, 2 and 3 were connected to boilers Nos. 1, 2 and 3 by the loop header *ABCD* with valves and drips as indicated.

Engine No. 3 was to be tested, and the first problem was to devise a means to accurately weigh the feed water used for steam in the engine alone. To accomplish this, boiler No. 2 was cut off entirely and boiler No. 1 was used to supply the engine. Boiler No. 3 was

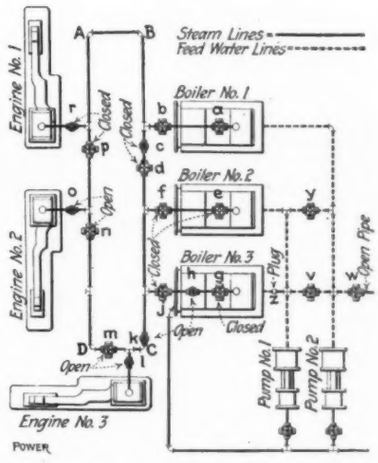


DIAGRAM OF THE ARRANGEMENT OF THE EQUIPMENT

used to supply the feed pumps and sump-pit ejector and to supply any other demand for steam. Engines Nos. 1 and 2 were shut down. All this was possible because the test was run on Sunday when the building was only partly occupied. Under these conditions it was necessary to isolate boiler No. 1 from the others by closing valves *g, j, e, f* and *p*. If it had been justifiable to assume that all the closed valves were not leaking, then it could also have been assumed that all steam generated in boiler No. 1 would have gone directly to the engine and that no steam would have gone to the header. Inasmuch as all the steam supplied to any part of the header came from boiler No. 1 and as this boiler was being fed with weighed water which was intended for the engine, it follows that if any steam had passed through leaky valves and then condensed, this weight of steam would have been charged to the engine and would have indicated a larger steam consumption than actually occurred. The problem therefore was whether any steam went astray, and, if so, how much. All doubt would have been removed if valves *j, d* and *p* could have been blanked.

The valves *n* and *m* were left open and the drips *k, l* and *o* were open and all condensation was caught in pails and returned to the feed-water barrel without being reweighed. That any condensation could pass through the valves *f* and *e* was out of the question, as whatever condensation there was would follow the line of least resistance and escape at the drip *k*. If any steam escaped through the valves *j* or *g* it would pass through the open drip *h*, but there was no leakage at this point. In this way the steam connections were proved to be tight and the condensation that did occur was reclaimed and accounted for.

The remaining observations and data were obtained without difficulty; diagrams were taken at frequent intervals. The back pressure, which was about ¼ lb., was measured by a U-tube connected to the exhaust line. The load was furnished by lights in the building and by a rheostat.

The next difficulty, after the main steam connections were taken care of, was the steam connection to the feed pumps. Under working conditions steam for the pumps was taken directly from the header, but during the test it was necessary to cut this off and a direct connection from boiler No. 3 had to be made. This was done by making a connection to the top of the water column. When steam passed out this way it was found that water was carried along from the lower part of the gage-glass and the pump cylinders knocked so hard that it was necessary to stop the pump at times. In order to overcome this the cock on the lower pipe to the water column was shut and the water level guessed at. As very little steam was drawn from this boiler this was permissible. However, when it was desired to know the water level the pump was shut down and the lower cock opened.

Thus all the difficulties in connection with the steam lines were overcome, but there was just as much question in regard to the feed-water connections. The lines were connected substantially as shown in the diagram. Pump No. 2 was used to supply boiler No. 1 with

water while pump No. 1 fed boiler No. 3. The question was whether any of the weighed water passed through the valves *v* or *y* and to boiler No. 3, or whether any of the feed water from pump No. 1 escaped into boiler No. 1. No water escaped into the idle boiler No. 2. It then remained to prove that *v* and *y* were tight in both directions. To do this pump No. 1 was stopped and pump No. 2 fed boiler No. 1. The clean-out plug *z* was taken out and it was found that no water escaped there. Then the plug was replaced, and the pump stopped; pump No. 1 was then started and the valve *w* was opened. If there had been any leakage in the valves *v* and *y* water would have come out of the open pipe outside of the valve *w*, but this did not occur and in this way valves *v* and y were shown to be tight; all difficulties were overcome and the test was run off in two four-hour periods.

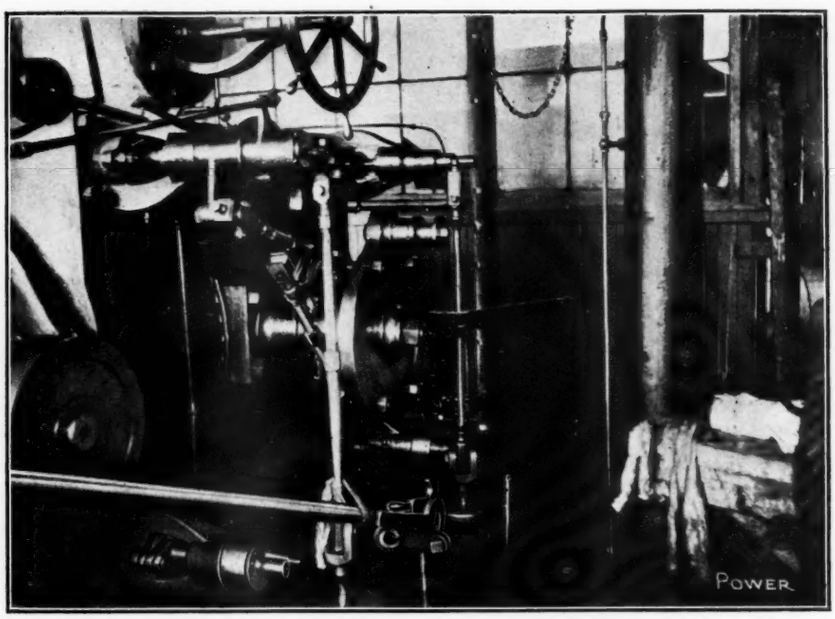


FIG. 2. SHOWING BROKEN VALVE GEAR

An Unusual Accident

BY H. E. OSGOOD

In shutting down a 32x42-in. single-eccentric Corliss engine running at 90 r.p.m. on 15 lb. pressure and 27 in. vacuum, water was drawn from the condenser into the cylinder which resulted in bringing the engine to a sudden stop. The arms of the flywheel were cracked, four at the hub and two at the rim, the main-bearing pedestal was demolished, the cylinder shifted ¾ in. on its base, the steam arm on the head end and the exhaust arm on the crank-end valves were broken, the piston rod was bent at the taper fit in the crosshead and the ¾x2½-in. key at the crosshead jammed ½ in. on the edge. The connecting-rod, crankpin and shaft sustained no damage, except for a light cut being necessary to smooth up the shaft at the main bearing, which was somewhat battered.

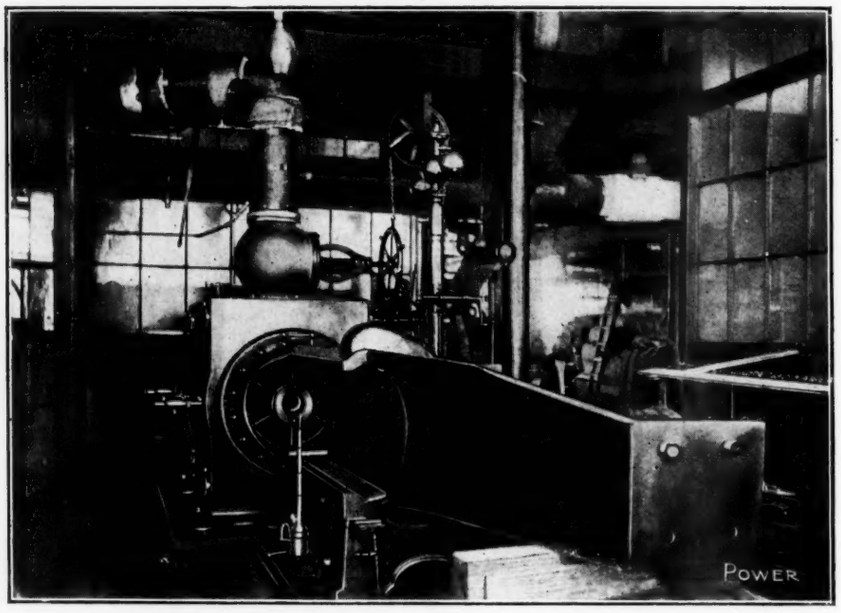


FIG. 1. ENGINE AFTER REMOVAL OF BROKEN PARTS

Fig. 1 shows the condition of the engine following the removal of the broken parts. The flywheel weighed 24,000 lb., was 16 ft. in diameter with a 36-in. face in two segments with four 2½-in. hub bolts and six 1½-in. rim bolts at each joint. There were six arms, each having 66 sq.in. of cross-sectional area at the hub and 45 sq.in. at the rim.

The accident resulted directly from water being drawn into the cylinder, before the engine had been brought to a stop. The steam was supposed to have been all but shut off, the clutch on the jackshaft was reported as having been thrown out, and, except for closing the exhaust valve, the engine was practically cut out of service.

Examination proved that the disk on the automatic check valve at the condenser had been removed some years

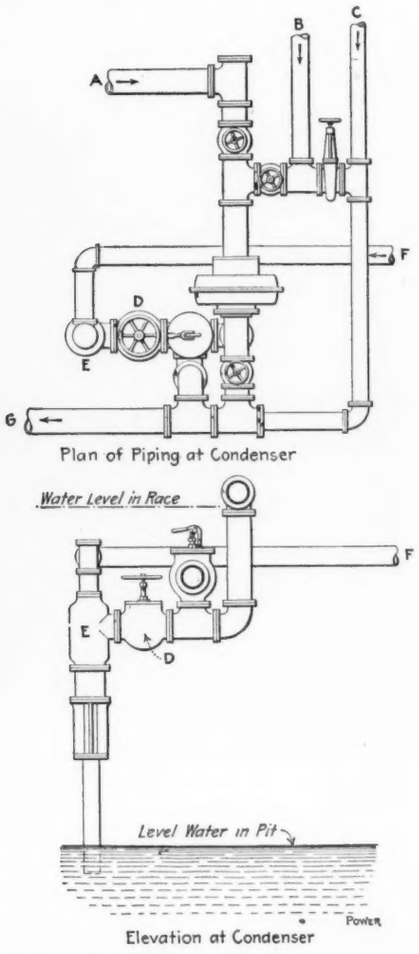


FIG. 3. CONDENSER PIPING

previous to the accident and never replaced; consequently, even if steam was entirely shut off, the momentum of the parts was sufficient to continue the motion, and there was nothing to prevent water from the condenser being drawn into the cylinder by the suction of the piston. The valves closed and the water thus entrapped brought the piston to a standstill, with the results as above described. Closing the main steam valve relieved the cylinder of any pressure to oppose an inrush of water. The water was drawn into the head end of the cylinder, as indicated by the compression of ½ in. of the key at the crosshead. It is surprising that the cylinder head was not broken.

Fig. 3 shows the exhaust-piping connections *A*, *B* and *C* from the 32x42-in. Corliss engine and Nos. 2 and 3 engines respectively and the location of the defective stop-check valve *D* and its relation to the condenser *E;* the water levels in the hotwell and also the race from which the injection water supply was taken through the pipe *F*. That both the race and hotwell are below the engine cylinder shows that the water was drawn into it by suction. The continuation of the exhaust line to the atmosphere is shown at *G*. The surplus water escaped while the exhaust valve was open on the return stroke, but the remainder formed a solid body resisting further motion of the piston.

Fig. 2 shows the valve mechanism, with the head-end steam-valve arm and the crank-end exhaust-valve arm, which were broken, removed. What caused these arms to break and the others not is an interesting point for speculation; the only likely supposition is that their valves were at the point of their most rapid movement, considering that the water was compressed in the head end, and the other valves were at a point of least motion, the momentum of the moving parts being sufficient to fracture the respective valve arms.

The Vibragraph

Vibrations in a power plant are always undesirable, but especially so when they occur in large urban power stations closely surrounded by other buildings. Although reciprocating engines are the chief offender, sometimes steam turbines also, owing to imperfect balance of the rotating parts, set up vibrations reaching to some distance.

The only instruments heretofore available for measuring vibration effect have been generally inconvenient to use, have left too much to the judgment of the observer, and have not been exact in tracing the source of the vibration.

Siemens Brothers & Co., London, have placed on the market the "vibragraph," which cuts out the personal equation and renders vibration visible in magnitude and direction. The vibragraph, Fig. 1, is a case inclosing the measuring apparatus proper. The vibrations are optically magnified so as to be clearly visible to the eye. A little mirror floating on a mercury drop and held against sidewise displacement throws a ray of light from

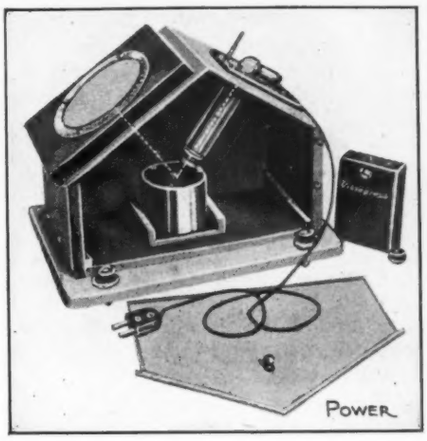


FIG. 1. THE VIBRAGRAPH

an incandescent lamp upon a sensitized paper or a ground glass. At complete rest the beam of light will form a point upon the paper, while a vibration in one plane will trace a straight line, and irregular and complicated vibrations curves or loops.

Each instrument has three measuring vessels of different sensitiveness, whose constants are laid out by the manufacturers in charts accompanying the apparatus so that a vibragraph leads to an

FIG. 2. VIBRAGRAM OF A TURBO-GENERATOR

easy determination of the actual magnitude of a vibration. Although the measuring vessels are very sensitive, they are so constructed as to need no special care in transportation. The apparatus ready for use weighs about 26 lb., and measures in its packing box about 14x10x8 in.

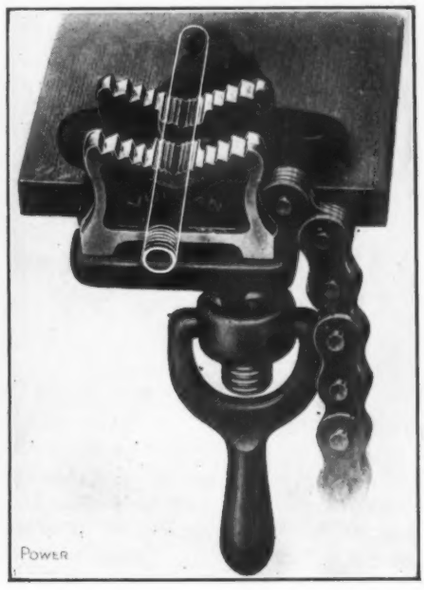
Fig. 2 shows the vibragram of a turbo-generator whose actual vibration is 0.005 in., and Fig. 3 the vibragram of a steam

FIG. 3. VIBRAGRAM OF A RECIPROCATING ENGINE

engine driving a 1250-kw. dynamo at 188 r.p.m., where the vibration amounts to 0.008 in.

Vulcan Chain Vise

Two new sizes have been added to the line of chain vises manufactured by J. H. Williams & Co., Brooklyn, N. Y. The larger size takes pipe ranging from ¾



VULCAN CHAIN VISE NO. 1

to 8 in. in diameter; the smaller has a pipe capacity of from ⅛ to 2 in. in diameter. The illustration shows the extended tooth which prevents bending or injuring the smallest pipe sizes when gripped in the vise.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Rotary Converters and Their Operation—II

Starting Rotary Converters

There are three ways of starting a rotary converter: by using a starting motor; by direct current, the converter starting as a shunt motor, and by alternating current, the converter starting as an induction motor.

The advantages claimed for the first are that heavy copper dampers can be used on the pole pieces, so proportioned that even under the most adverse conditions hunting is prevented. The small starting current required avoids any disturbance on the alternating-current system and when necessary all the converters can be started simultaneously.

When direct current is available the converter can be started as a direct-current shunt motor, and switched in on the alternating-current lines when it is in synchronism.

Rotary converters can also be started from the alternating-current side but this is not recommended except in special cases. Like the synchronous motor, it only develops a normal torque when in synchronism. To obtain the best operating performance the converter should have heavy copper dampers; but these interfere with starting on alternating-current without a starting motor, as it greatly increases the starting current. The power factor of the converter, while starting in this way, is low; and this, with the heavy current drawn from the line, causes flashing at the commutator. Also any such sudden demand for current will affect the voltage regulation of the supply circuit.

Where conditions likely to cause hunting are absent, self-starting will much less disturb the supply circuit if less effective dampers of smaller sections are used.

The starting motor is usually of the induction type having two poles less than the converter, and mounted on the armature shaft of the latter just outside of the bearing. This enables the motor to bring the converter up to and above synchronous speed and regulating devices are arranged for accurate synchronizing. Four-pole rotary converters usually have alternating-current, series-wound, commutator type starting motors, as there is too great a difference between its speed and the speed of a bipolar induction motor.

After a converter has been in service a short time the friction of the bearings will decrease, and the starting motor speed will have to be readjusted, or it will over-run the synchronous speed. Bearing friction also varies with changes of temperature, lubrication and other conditions impossible to accurately control. A resistance can be inserted between the alternating-current terminals of the converter, which will act as a load upon the starting motor, to check its speed and secure synchronism. When the converter is connected to the line this synchronizing rheostat must be out of circuit.

When a converter is started from the direct-current side or by a motor it must be brought into synchronism with the alternating-current supply before being connected to the line. Lamps are often used for synchronizing, but are objectionable because only a considerable difference of phase makes them glow and they do not indicate whether the converter speed is too fast or too slow. The ideal synchronizer should show the latter, the difference in speed, and the exact time of synchronism; these requirements are met by the synchroscope. The automatic synchronizer is an instrument designed to eliminate the uncertainty and trouble which occasionally arise when hand-thrown switches are used and insures that no machines will be thrown on the system unless conditions are right. When either a starting motor or direct-current starting is used, self-synchronizing can be obtained by closing the switches on the alternating-cur-

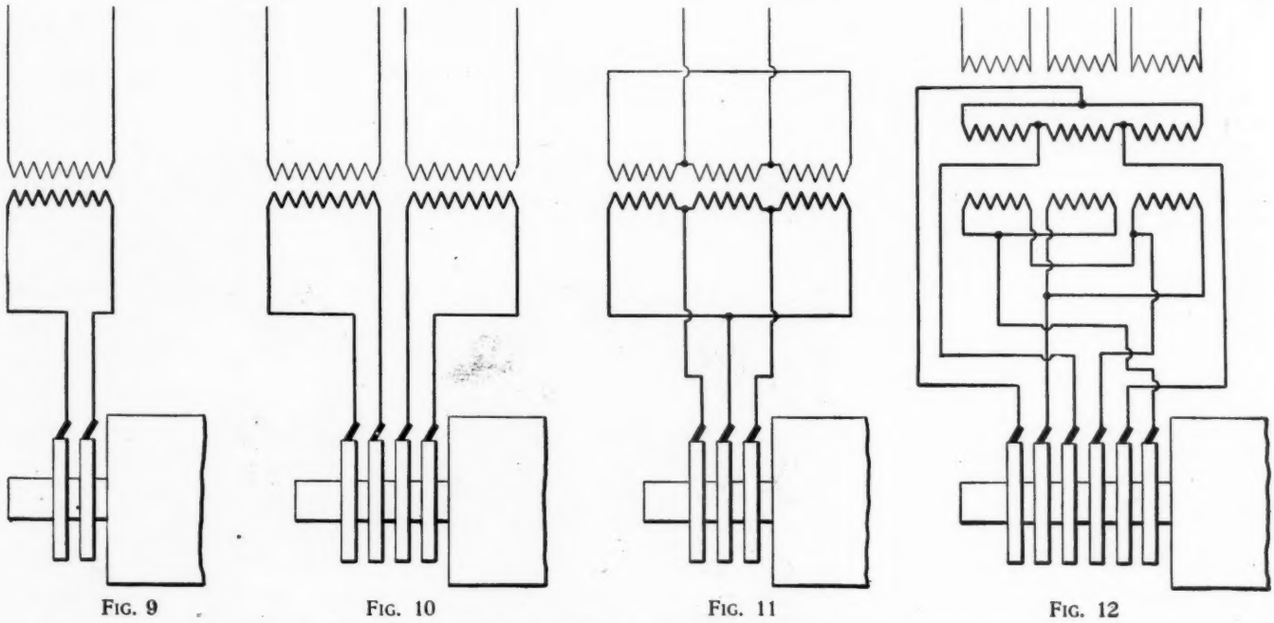


Fig. 9 Fig. 10 Fig. 11 Fig. 12

Transformer Connections for One-, Two-, Three- and Six-Phase Rotary Converters

rent side through proper reactance coils when the converter is above synchronous speed. The converter will lock in step as it passes through synchronous speed.

Parallel Operation of Rotary Converters

If several converters are to supply the same direct-current system they can be connected in parallel like shunt- or compound-wound direct-current generators. With compound-wound converters an equalizing connection must be used. The load will be divided between the different converters according to their direct-current voltages. The voltage can be regulated by changing the alternating-current voltage; or by regulating the field current, which changes the direct-current voltage by increasing the drop in voltage in the armature; or by shifting the direct-current brushes.

It is not advisable to have rotary converters, supplying the same direct-current system, connected on the alternating-current side; that is, supplied from the same transformer secondaries. This forms a closed local circuit in which heavy cross currents may be established and makes impossible adjusting the alternating-current voltage so as to compensate for the slightly different voltages of the several converters.

Rotary converters at different stations can be used to feed into one set of mains, if their voltage is properly adjusted. The direct-current sides can be handled almost exactly as independent direct-current generators, except that so wide a variation of voltage cannot be secured by adjusting the field current. Two rotary converters taking alternating current from two separate secondary circuits can be operated in series on a direct-current three-wire system. A single converter taking current from a pair of transformers can also be operated on a three-wire system, the neutral wire of the direct-current system being taken from the interconnected point of the transformer secondaries.

Oscillators and Overspeed Devices

The armature of a horizontal rotary converter will normally take a fixed position relative to the field frame and revolve without end movement. When running in one position the brushes are likely to wear grooves in the commutator and collector rings, and the bearings do not operate favorably. The oscillator is designed to produce a periodic longitudinal motion of the shaft, and may be either mechanical or magnetic, depending upon the source of the required impulse.

Where rotary converters may be operated at times from the direct-current side, running away due to a weakened field should be guarded against. This can be done by special individual exciters or a spring-closed knife switch operated by a centrifugal tripping device.

Inverted Rotary Converters

When driven by direct current and supplying alternating currents, the speed of a converter is not held constant by the alternating-current circuit, but, like a direct-current motor, is governed by the field strength and, therefore, by the armature reaction. If the alternating load is inductive, changes of load may cause wide variations in speed and frequency.

Inverted rotary converters have a special exciter, directly connected to the converter or driven by an induction motor receiving current from the alternating-current side. This exciter has an unsaturated magnetic field so that its voltage is considerably increased by a slight increase in speed. The resulting increase in the exciting voltage of the converter produces a stronger field and prevents the speed from being unduly increased.

Figs. 9 to 12 show the connections of substation transformers for single-, two-, three- and six-phase rotary converters respectively.

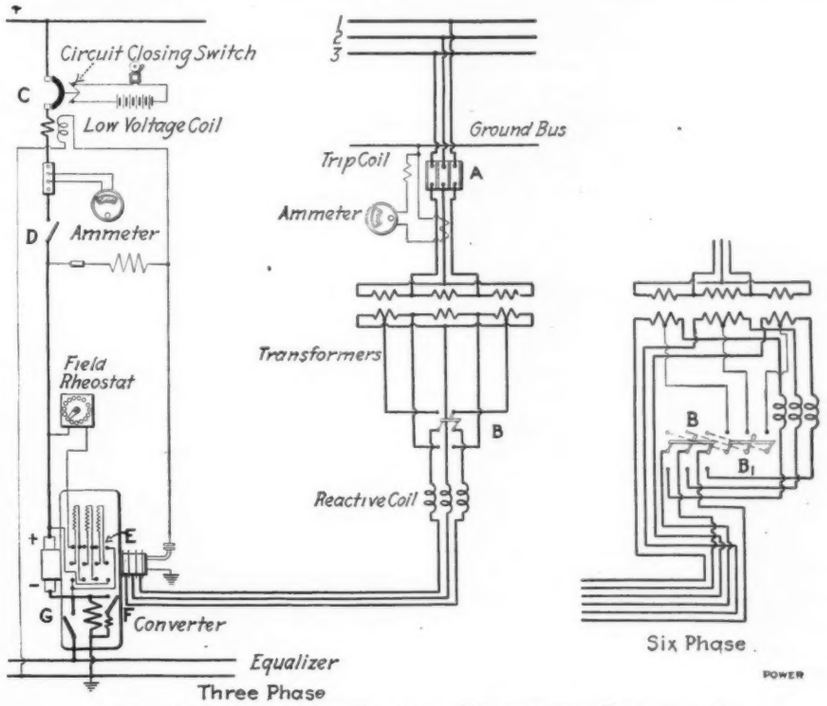


Fig. 13. Starting and Running Circuits for Three- and Six-phase Converters

Operating Hints

On heavy loads best results will be obtained with the brushes set at a slight lead from the neutral position, which can be determined by trial. To prevent sparking when starting from the alternating-current side, the brushes usually have to be shifted back to the neutral position. Occasionally a machine will not start from the alternating-current side if the voltage is low. This can often be overcome by shifting back the rocker-arm. Bolts should be occasionally tightened on the field winding connections, brush-holder studs and similar points. A good quality of oil should be used in the bearings and care should be taken to use a heavy oil on large machines. Direct-current brush springs should be set in the first notch if the commutators are in perfect condition; otherwise in the middle notch for ordinary loads. The speed-limiting devices should be tested at regular periods.

Using compressed air to blow out internal passages is strongly urged. Particular attention should be paid to the spaces under the collector rings; copper cuttings must not be allowed to accumulate in grease or oil around these parts.

Grinding or turning commutators should not be required if they are carefully attended to, unless heavily used or unless some fault develops such as a flat or high bar. In case the commutator condition becomes bad, grinding has certain advantages over turning; it removes less material, requires less skill and leaves the commutator in better condition.

In turning, care should be taken to block the shaft to prevent end play, and the speed should be as high as possible without burning the tool; this is usually not above 300 ft. per minute. Before turning the commutator, means should be provided to prevent chips or dust from working into the armature. After turning, the end of the commutator should be rounded off with a file to a radius of at least $\frac{1}{8}$ in. The commutator should then be examined bar to bar to see that no small particles of copper are em-

bedded in the mica. Under no circumstances should emery be used for grinding or polishing a commutator.

Switching

The switch gear supplied with converters varies so much that complete instructions, covering all cases, cannot be given. The following applies to the usual connections and to the switchboard equipment of compound-wound converters in railway service. Referring to Fig. 13 a machine should be started as follows, assuming that all switches are open before starting and that the plug is inserted in the potential receptacle on the direct-current switchboard panel to connect the machine to the voltmeter:

Close the main high-tension switch *A*, then the starting switch *B* upward. For six-phase machines both switches *B* and B_1 should be closed upward. The machine will then run up to speed, which will be indicated by a cessation of the beats of the direct-current voltmeter. Close the equalizer switch *G* and the series-shunt switch *F*. Close the switch *E* into the top position. Throw the starting switch *B* quickly from the top to the bottom contacts. Adjust the direct-current voltage to approximately that of the busbars. Push up the low-voltage release of the circuit-breaker and close th circuit-breaker *C*. Close the main switch *D*. Adjust the division of the load between the machines, if more than one are in service, by means of the field rheostats.

To shut down a rotary converter, open the circuit-breaker *C*, pull out and turn the circuit-closing auxiliary switch to stop the ringing of the alarm bell; open the main switch *D;* open the high-tension oil switch *A;* allow the machine to retard until the voltage falls off to about 100 before opening the field switch *E* or the starting switch *B;* open the field switch *E*, equalizer switch *G*, series-shunt switch *F* and starting switch *B*.

If other machines are carrying load when a compound-wound converter is started, the correct polarity may be insured by closing the equalizer switch *G* when the machine locks in step. By watching the swings of the direct-current voltmeter as the machine approaches synchronism, switch *G* may be closed just previous to the last two or three swings, thus insuring proper locking on the first trial, if there is current for the series field from the other machines. If the machine locks with the wrong polarity as indicated by the direct-current voltmeter needle going down off the scale, the field switch *E* must be closed first into the down position, which will cause the voltmeter to return above zero, when the switch *E* must be pulled out and closed into the top position. The reversal of polarity should be made while the machine is running on the first starting tap.

Do not open the field switch *E* until the machine voltage has run down; otherwise the arc on opening the inductive field circuit may be blown by the armature blast from point to point, causing a short-circuit, or the strain may injure the insulation of the field winding.

Do not open the starting switches until the voltage has decreased to zero. This leaves the transformers without a heavy residual magnetism, which tends to cause an extra rush when they are subsequently thrown into circuit.

Do not open the oil switch with the converter on the starting tap and the field closed; otherwise the converter may build up to full voltage, tending to run up the transformers to considerable above normal voltage.

Do not parallel the machines on the direct-current side without closing the equalizing switches.

Do not start the converters with the field switch *E* closed either in up or down position or with the series-shunt switch closed.

Do not close the alternating-current starting switch slowly into the top contacts, for if this is done, the rush of current may pit the tips of the clips and thus prevent the switch blades from going in.

Do not close any switch slowly.

Do not close the circuit-breaker with the main switch *D* closed.

Do not close the circuit-breaker after a heavy short, without plugging the voltmeter to the converter to make sure that the polarity is not reversed.

LETTERS

Reversed Polarity

Our plant is equipped with two 125-volt, 720-ampere direct-current dynamos wired to be run in parallel. A few days ago the electrician stopped No. 2 and gave it a good cleaning, also sandpapering the armature and brushes. When ready to start it, he threw in the circuit-breaker and the three-pole switch, and told his helper to start the dynamo. The latter started to do so very carefully, but before the machine had attained full speed the electrician threw in the paralleling switches.

The circuit-breaker immediately flew out and the voltmeter and ammeter on No. 2 dynamo tended to go backward. We changed the meter wiring accordingly and they indicated all right.

We again tried to run the dynamos in parallel, when both circuit-breakers came out with a bang. We then decided that the polarity had been reversed and the question was how to change it back again. The master mechanic came to our aid and ran two wires from No. 1 machine to the field connections of No. 2, placing a cutout with 30-ampere fuses in circuit as a precaution. This changed the polarity back again. We then changed the meter wires back to their original connections, started the dynamo and threw in the paralleling switches and they ran all right.

James Mitchell.

North Adams, Mass.

Home Made Welding Device

Upon taking charge of our shop, I found a number of broken grids in the resistance for the crane controllers. The broken parts had been connected by splicing copper wire on each side, but this was very unsatisfactory, as they became loose and began to burn. I removed the broken parts and repaired them with a home-made electric welding device. This was made by connecting some No. 4 wire to the crane circuit, the positive wire to an old resistance, and the negative wire to the carbon holder. From the resistance the wire led to a plate upon which was placed the grid to be welded.

The carbon holder is made of ¾-in. pipe holding the wire on one end and two clips which hold ¾-in. carbon blocks on the other end; a handle was placed on the end to which the wire is connected.

Striking the arc at the brake and keeping it moving back and forth, it took but a few seconds for the metal to weld. Then feeding a little of the same metal in while moving the arc back and forth, I soon had the grid a little thicker at the fracture, repeating the same operation on both sides. It is now as good as new and has been in use several months. I protected my hands with a pair of gloves and my face with ⅛-in. ruby glass.

C. A. Blomberg.

Detroit, Mich.

What Made Transformers Heat Up?

We had occasion to disconnect one of our switchboard wattmeters, and left the secondaries of the potential and current transformers in open circuit. These transformers were operating on a three-phase 2300-volt circuit and I noticed after one night's operation that one of the current transformers became so hot that some of the insulating compound ran out.

I therefore put a jumper across the transformers and the next night that one ran cool but the other became hot again. I then did likewise with the second transformer and ran both with jumpers until the meter was back on the board. Why did the current transformers get hot while running with the secondaries open?

C. J. Brindley.

Lorain, Ohio.

Before July 1, the National Electric Light Association hopes to have 12,000 members. There were 11,732 on May 15.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

Small Gas Power Plant

By A. R. Maujer

Unless, for some special reason, the central station can and will sell current at or very close to actual cost, or unless the load factor is low, it is usually possible for the consumer to generate power for less than it would cost to purchase it. This is fairly demonstrated by the following description and operating data of the plant of Frank Holton & Co., 2638 to 2646 Gladys Ave., Chicago, Ill., makers of band instruments.

The factory is three stories high and covers a plot 86x100 ft. It is heated by a low-pressure gravity-return heating system, steam for which is generated in a small locomotive-type boiler.

The numerous lathes, drill presses, buffing wheels, etc., are driven in groups from motor-driven jackshafts. The motor load is about 40 hp. and in addition there is a lighting load consisting of about 200 incandescent lamps. To supply the original requirements a 15-kw. direct-current dynamo belted to a 25-hp. two-cylinder Nash gas engine was installed in 1907 (see Fig. 1). By 1911 the load had grown beyond the capacity of this unit and the equipment shown in Fig. 2 was installed. This consists of a Crocker-Wheeler 220-volt direct-current dynamo of 25 kw. rated capacity driven by a 40-hp. Nash gas engine.

The engine is of the two-cylinder type, operating on the four-stroke cycle, equipped with a throttling governor, and running at 275 r.p.m. Ignition current is supplied from the small storage battery seen on the side wall in Fig. 2. This battery is kept charged by a special igniter generator, mounted on the frame of the engine and driven by a belt from the main shaft. Illuminating gas is used as fuel.

The engine is started by compressed air at 180 lb. pressure, a small motor-driven single-cylinder vertical compressor being used to store air in two small steel tanks.

After passing through the cylinder jackets the cooling water is run through a transverse-current water heater made by the Williams Tool Co. which utilizes the heat of the engine exhaust to raise the temperature of some 200 gal. of water per hour from about 140 to 180 deg. F. All of this water is used for manufacturing purposes.

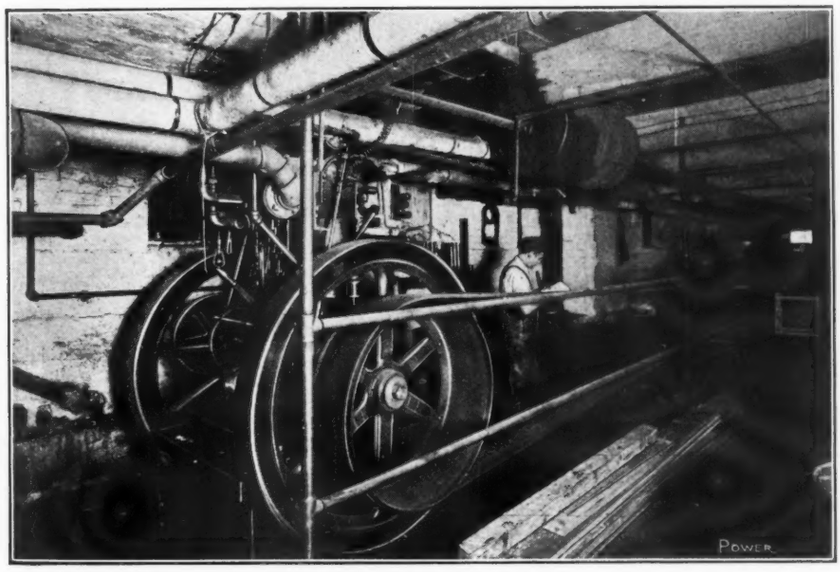


Fig. 1. Original Installation

Operating Costs

Table 1 gives the gas consumption and electrical output for November, 1911, which is a typical month. The engineer, in addition to looking after the generating equipment, runs the heating plant and takes care of all the motors, shafting, wiring, piping, plumbing, etc., throughout the building. Even if the generating apparatus were not installed his services or their equivalent would be required 75 per cent. of the time; hence, in estimating the cost of power, but 25 per cent. of his salary, which is $100 per month, can properly be included. As all of the cooling water is used throughout the factory, the cost of the water is not chargeable against the engine. In fact, this water would have to be heated by other means and hence the engine really should be credited with this item. The average consumption of lubricating oil is 0.9 gal. per day and costs 36c. per

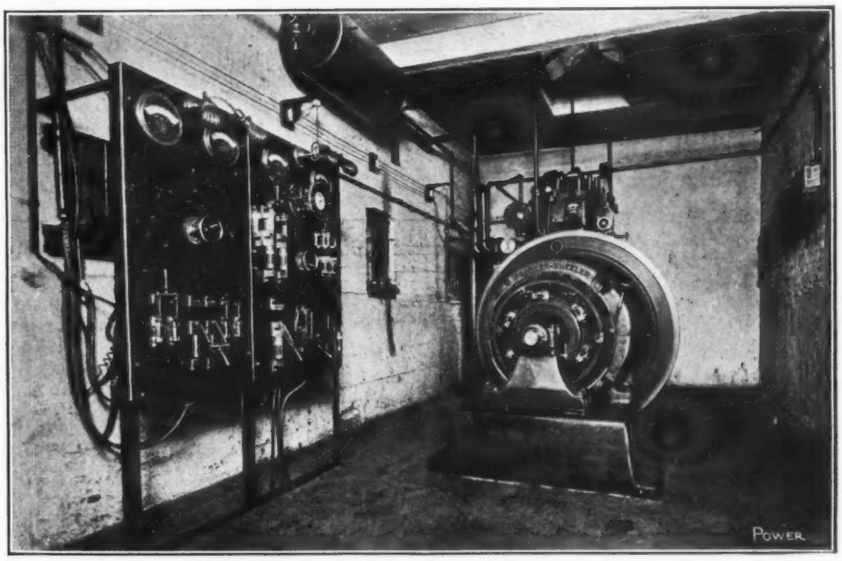


Fig. 2. 25-kw. Unit and Switchboard

gallon. At this writing, a year after installation, there have been no repair charges against the power equipment. The cost of the 25-kw. unit with all accessories installed complete was $3250 and that of the 15-kw. unit was $1775.

Based on the foregoing consideration and on the data given in Table 1, the following estimate of the cost of power for Nov. 1, 1911, is fairly accurate:

Fuel (121,100 cu.ft. of gas@80c. per 1000 cu.ft.)	$96.88
Labor (25 per cent. of engineer's time)	25.00
Oil	8.10
Interest, depreciation, etc. (12 per cent. per annum on $5025)	50.25
	$180.23

$$Cost\ per\ kw.\text{-}hr. = \frac{\$180\ 23}{4542} = 3.97c.$$

Cost of Heating

The heating does not affect the power cost one way or the other as it is exactly the same regardless of whether the power is generated in the building or purchased from some outside source. It may, however, be interesting to know what this cost is. During the winter of 1911-12 the heating plant consumed 54 tons of Pocahontas coal costing $3.75 per ton and 108 tons costing $4, making a total fuel cost for the heating plant of $634.50.

TABLE 1. DAILY FUEL CONSUMPTION AND ELECTRICAL OUTPUT FOR NOVEMBER, 1911

Date	Gas Consumed, Cu.Ft.	Output, Kw.-hr.	Fuel Cost per Kw.-hr., Cents
1	5,200	192	2.16
2	5,300	193	2.20
3	5,100	188	1.27
4	2,900	106	2.19
6	5,000	179	2.24
7	4,900	186	2.11
8	4,900	184	2.13
9	5,100	186	2.19
10	4,900	184	2.13
11	3,000	105	2.29
13	5,000	198	2.02
14	5,200	202	2.06
15	5,100	195	2.09
16	5,100	194	2.10
17	5,600	196	2.29
18	3,100	116	2.12
20	5,400	204	2.12
21	5,300	205	2.07
22	5,500	207	2.12
23	5,400	207	2.08
24	5,600	206	2.18
25	3,200	124	2.06
27	5,400	205	2.10
28	5,000	195	2.05
29	4,900	185	2.12
Total.	121,100	4542	2.13

Fuel Supply Troubles

By James H. Beattie

An engine, to give steady, uninterrupted service, must have constant fuel supply; furthermore, the fuel must be clean. Greater care is necessary perhaps to keep an engine using liquid fuel in constant operation than one using gas, as the former is more subject to trouble through dirt or water in the fuel, than an engine using natural or artificial gas as these are reasonably constant in their composition and not, as a rule, subject to interruption in flow. The following instances may be of interest as they are good examples of the kind of difficulties the average engine operator meets.

A small vertical engine suddenly developed a bad case of backfiring, misfiring and loss of power. The ignition system was in good condition and the spark properly timed. Opening the needle valve (the engine operated on gasoline) did not improve matters. Throttling the air pipe helped somewhat but did not effect a cure. A section of the cylinder and mixing valve of this engine is represented in Fig. 1. The mixing valve of the common overflow type is indicated at *A*, the gasoline being kept at a constant level by the overflow pipe. On the suction stroke of the engine the gasoline is drawn through the small pipe *C*, the amount entering being determined by the setting of the needle valve. Dirt had entered the small nozzle and lodged near the end where the mixture of the fuel and air occurred. The pipe was not completely stopped up but very nearly so; hence the amount of fuel that could pass was regulated by the obstruction and not by the needle valve. Throttling the air supply increased the suction and thus more fuel could pass the obstruction in a given time. The whole trouble disappeared when the obstruction was removed. Lack of care in filling the fuel tank was responsible for this difficulty, which goes to show that too great care cannot be taken to keep the fuel free from dust, dirt or water.

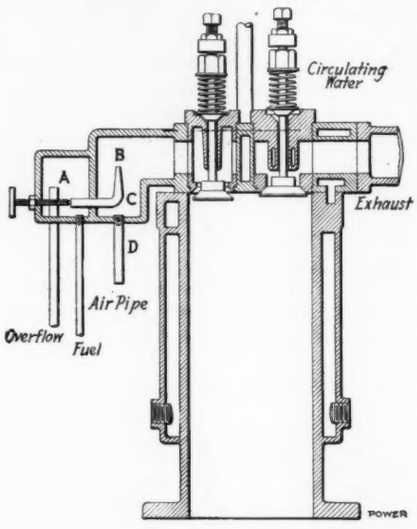


FIG. 1. SECTION THROUGH CYLINDER

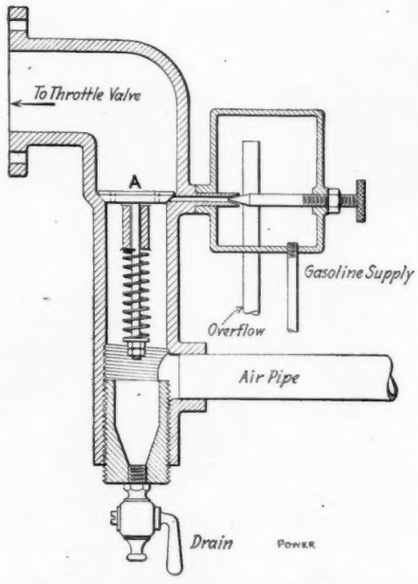


FIG. 2. SHOWING MIXING CHAMBER

Another engine operated very satisfactorily for several weeks after it was installed, and then began stopping frequently, for no apparent reason. It proved to be another case of fuel-supply failure, this time due to the fuel tank being too far from the engine and too far below the gasoline pump. While the pump was new and in perfect condition it brought the fuel to the engine without trouble, but when the packing began to wear the pump was not equal to the task. Pumping by hand before starting the engine would bring up enough fuel to keep the engine going for a short time, as the mixing valve held considerable, then the engine would stop again. The trouble was remedied by placing the fuel tank nearer the engine and at a higher level.

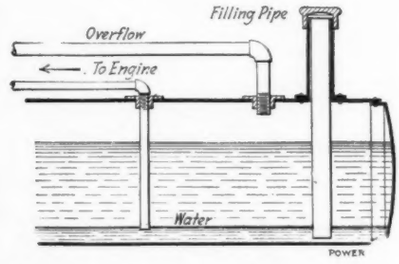


FIG. 3. SHOWING WATER IN BOTTOM OF TANK

In another case a small engine was rapidly getting a bad reputation by using more fuel than necessary. It was rated at 15 hp. and consumed nearly 25 gal. of gasoline per day, although not operating at full load. The trouble proved to be looseness in the lock lever that holds the inlet valve shut when the engine is coasting. This caused the inlet valve to chatter (due to the partial vacuum in the cylinder when the engine was running idle) allowing fuel to enter and go to waste. The fuel consumption dropped to about 12 gal. per eight hours after proper adjustments were made.

The field man frequently gets more calls to imaginary troubles than to real ones. One man who had just purchased an engine wrote the general agents that the engine would not run and to send a man at once. If the trouble proved to be due to the engine the company agreed

to pay the expenses of the trip, but if the fault was the owner's, he was to pay them. The pipe leading from the supply tank to the carburetor extended about half an inch into the tank so that any dirt which might be in the tank would not get into the carburetor. There was only enough fuel in the tank to cover the bottom, its level being below the end of the pipe. The man was sure he had fuel in the tank for he could see it. After 5 gal. of gasoline had been placed in the tank and the engine started he paid the bill without a murmur.

In a similar case the fuel tank was outside the building, and, although there was plenty of fuel in it, a cock in the pipe to the engine was closed. This man had been trying for two days to get his engine started.

Fig. 2 is a cross-section of the carburetor and air valve of a throttle-governed engine operating a small electric-light plant. After operating satisfactorily for a day the engine refused to start the next evening. As will be noticed from the sketch, the opening from the carburetor to the port leading to the throttle valve is covered by the edge of the disk *A*, which is held on its seat by a light spring. The disk is perforated so that air may pass through whether the opening from the needle valve is open or not. This valve was stuck on its seat, the result being that the engine could get no fuel.

Another puzzling difficulty was with an engine employing a pump feed for the gasoline. This engine was served by a tank such as illustrated in Fig. 3, the pipe to the fuel pump extending to within an inch of the bottom of the tank. Water had accumulated in the bottom of the tank until it had reached the end of the pipe, and the pump could get nothing but water; hence a stop with considerable difficulty resulted before the trouble was remedied.

The Gas Turbine

Zeitschrift des Vereines deutscher Ingenieure for Mar. 30 of the current year prints a contribution by Dr. A. Stodola, of Zürich, in review and criticism of a recent book by Hans Holzwarth on the theory and construction of the gas turbine, together with operating results obtained from two of that type.

He says that in the Holzwarth design the working cycle is a periodic release and explosion of a predetermined quantity of fresh charge which then, under the effect of expansion, is directed through a nozzle to the rotor of the turbine. When the expansion has continued to nearly the back pressure, air is admitted to the explosion chamber until the latter is scavenged of the burned gases and the temperature is lowered sufficiently. The nozzle valve is then closed and gas or a gas-air mixture under pressure is injected into the chamber for explosion and a repetition of the process.*

The inventor decided on compression of the gas only, whereas the air is supplied to the chamber by exterior pressure and to this end an exhaust fan maintains in the turbine a vacuum of about 3 in. of mercury. As the compression in the explosion chamber rises only to about 18 to 25 lb. the motor belongs to that class of explosion turbines having only very low advance compression; that is, compression for the fuel delivery into the combustion chamber. The designer chose the almost insignificantly low compression of 1.5 to 2 atmospheres because he endeavored to offset the necessary power requirement by the recovery in a regenerator of the heat of the gases of combustion, for which purpose these are conducted to a vertical tubular boiler and the steam thus generated supplies a steam turbine operating the centrifugal compressor and the exhaust fan. A theoretical calculation shows that this energy requirement amounts to 10 or 15 per cent. of the available duty of the turbine, which, calculated as heat, certainly means 35 per cent. of the calorific value of the gas used.

Notwithstanding the considerable heat recovery from the exhaust gases at relatively low temperatures the required heating surface is not impracticably large, but rather comparable to the cooling area of a surface condenser for a steam turbine of like duty.

In so far as the described arrangement is concerned, the Holzwarth turbine does not differ from well known designs, whereas its behavior in the matter of temperature shows a peculiarity of design. Thus the designer absolutely excludes all cooling by either jacket or spray and intends to accomplish the required reduction of combustion chamber and bucket temperatures by means of the scavenging air alone. Professor Stodola says: "In the face of this proceeding, one is involuntarily reminded of Diesel who also sought to build a motor devoid of cooling but who after years of battle against it had to accept the water jacket as a necessary evil." He says further that it is deserving of thanks that the Holzwarth book frankly presents and critically speaks of all the unfavorable experiences and the disillusionments of the early investigations.

Originally the gas-air reservoir was built around the combustion chamber as a jacket and the fresh change was thus heated before its admission. No wonder that there was early or pre-ignition.

In explanation of the great difference observed between the theoretical and the actual duties the author brings forward the difficulties of scavenging the chambers, inasmuch as the jet of gas issuing from one chamber interferes with the free exit of the air from the neighboring one, although the gap had been materially widened and a bypass around the buckets provided for the scavenging air. Generally, however, it seems that the nozzle is to blame as it does not yield a sufficient drop in the pressure of explosion, and thus the issuing jet may not expand to the pressure induced by the exhaust fan and the expansion established outside of the nozzle, in the widened gap, may not operate to the benefit of further acceleration of the jet.

During all the tests unusually low service demand was made on the turbine and it is evident that a 10 to 15 per cent. load requires less cooling.

The admissible temperature for the buckets at full load remains the main question with the gas turbine and this the investigations leave unanswered. It thus seems idle to make conjectures as to its ultimate efficiency, for only in the light of this can a determination be made of its economy as compared with the reciprocating gas engine and the no less serious competition of the steam turbine.

Though the gas turbine problem has not been solved by the Holzwarth design, Professor Stodola acknowledges indebtedness to it for valuable disclosures as to numerous details of the development. Of particular prominence is the series of tests of the first turbine relative to the use of the heavier fuel oils, including coal-tar oil.

In a word, Professor Stodola says that the Holzwarth investigations are a powerful incentive to renewed activity in the direction of the gas turbine, for it is and will continue to be an ideal of the designing engineer.

*See Power, Feb. 6, 1912.

LETTER

Igniter Points

For the make-and-break igniters on our producer-gas engine we had been using points that cost 25c. apiece. This meant $1.50 for points for each set of three igniters, so I began looking for a substitute. After using various metals I tried a brass point in the terminal rod together with a cold-rolled steel point in the igniter stem (or movable stem), and have been getting good service from this combination. For this purpose I turned down some old ¼-in. brass rods to the proper diameter for the points and sawed off pieces as they were wanted, filling a taper on one end to drive into the hole of the igniter terminal. I did the same with the cold-rolled steel points for the igniter stem.

With both points of cold-rolled steel the igniter would miss when running on the batteries, but would run all right direct from the battery-charging generator. However, as that gave such excessive ignition current it caused backfiring.

J. W. FRIES.

Middleboro, Mass.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Automatic Refrigeration an Ideal Central Station Load

BY PORTER L. SWIFT

Supplying electric current for the operation of automatic electro-mechanical refrigerating plants furnishes one of the largest and most profitable fields for the extension of electric service into commercial use. From the viewpoint of the progressive central-station manager, it is a most desirable addition to the load on lighting mains. The present perfected apparatus will produce no undue line disturbances through the automatic starting and stopping of the motor.

The reasons for the foregoing statements are numerous and self-evident if one will prepare a small chart divided longitudinally into 12 equal divisions representing the months of the year; at the left divide vertically for the degrees of temperature as on an ordinary thermometer scale and at the right divide vertically for the number of hours of darkness per day, which will range from about seven in the summer to nearly fifteen in the winter. On this chart plot the temperature curve for any locality for the entire year and the hours of darkness per day for the same period. The refrigerating duty will follow the temperature curve and the lighting load the hours-of-darkness curve.

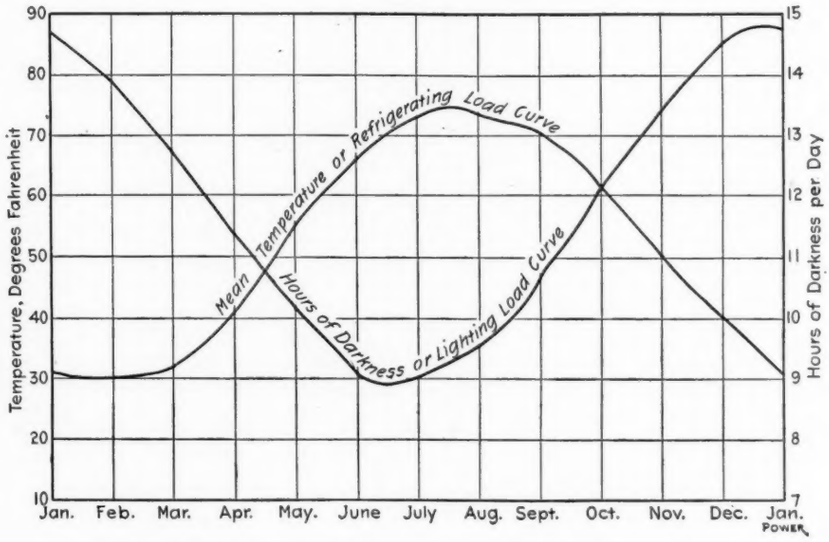
Herewith is a chart for New York. It may be seen that the dark periods range from a minimum of 8 hr. 55 min. in June to a maximum of 14 hr. 42 min. in December. These figures are from sunset to sunrise and the actual dark period in summer is shorter on account of the long light periods before sunrise and after sunset. The temperature curve is the average curve, while the actual refrigerating load will follow the maximum and minimum curve each day. The curve shows an average temperature around 75 deg. F. during July and August, while the maximum temperature for several days reaches 85 deg. and even 100 deg., and in this time there are only six to eight hours of actual darkness. During January and February temperatures as low as 20 deg. are reached for several days when lighting requirements extend 14 hr. or more.

A refrigerating load is the exact reverse of a lighting load, both in the months of the year and the hours of each day. It is heaviest in summer when the lighting requirements are small, and it is a long load, especially with automatic refrigerating plants, when the operation may reach 18 to 24 hr. each day. It is a day load at all times during the year because in the daytime the cooled supplies are removed from the refrigerators and replaced by new supplies. At the time the electric-lighting load is increasing it becomes less because the refrigerators are usually closed and well cooled during the later afternoon.

It would appear then that this class of power consumer ought to be encouraged by a low rate. In one Eastern city of 100,000 inhabitants the power company has found it advantageous to make a reasonably low flat meter rate for its refrigerating business. It was supplying some 21 of these refrigerating plants, last year realizing over $12,000 from this class of service alone. More plants are being installed and the management contemplates still further reducing an already low rate to encourage the installation of refrigerating plants, the majority of which are automatically controlled and driven by electric motors ranging from 2 to 25 hp. This business is nearly clear profit as the peak load for this service does not add to the general peak but follows it and comes at a time when the valley period is greatest. A load that is off the peak is a good one to cultivate, and in order that commercial departments may work up this new business a few words for the automatically controlled and electrically driven refrigerating plant may be of interest.

Any refrigerating load is desirable, but the automatic refrigerating load is doubly so because the automatic plant derives its motive power solely from an electric motor and for this reason alone is worthy of first consideration. The ordinary small refrigerating plant is usually operated during the daytime only and sufficient refrigeration is stored in brine tanks to hold the temperature in the refrigerators within the desired limit; but in extremely hot weather uniform temperatures are difficult if not impossible to maintain. The automatic type of refrigerating plant with its thermostatically controlled motor is available for operation during the whole 24 hr. and will maintain an even temperature in the refrigerator. This feature is especially attractive from a central-station viewpoint in that smaller motors are required to operate the automatic plants; this results in the longer use of a smaller maximum demand than is obtained with the ordinary type of machine, and in turn requires a smaller central-station investment to meet the consumers' demands.

There are few central-station loads which will show a better return per connected horsepower than by motors driving these automatic plants, for it is the rule that this plant will do the work with a 5-hp. motor which a nonautomatic plant will not do as satisfactorily when driven by a 10-hp. motor.



MEAN TEMPERATURE AND LIGHTING LOAD CURVES FOR NEW YORK

Automatic refrigerating plants are now built having a refrigerating capacity equivalent to the melting of 700 lb. to 16 tons of ice in 24 hr. and they are driven by motors, using either direct or alternating current ranging from 1 to 30 hp. One plant in New York City operates two of these 16-ton units driven by two 30-hp. direct-current motors.

The devices which control the operation of the automatic plant and their functions may be of some interest. The automatic expansion valve is so constructed that, for any given temperature to be maintained around the expansion coils, there will be within the coils a density of vapor proportioned to and governed by the pressure of the vapor itself. This gives a better regulation of the ammonia supply to the expansion coils than is possible with the closest personal attention and a hand expansion valve.

Another type of automatic expansion valve is so constructed that the temperature of the returning vapor from the expansion coils acts on a given volume of liquid and gaseous ammonia confined in a chamber which is connected with a diaphragm controlling the action of the expansion valve. The expansion or contraction of this confined liquid in the chamber, caused by the varying temperature of the vapor from the expansion coils, exerts a pressure on the diaphragm which controls the feed to the pipe coils. Thus the supply of ammonia to the refrigerator coils is governed by the temperature of the suction gas from the same coils after the heat in the refrigerator is absorbed, and the supply will be throttled and cut off entirely as conditions in the refrigerator require. The frost line will also be prevented from reaching the ammonia compressor.

The automatic water valve is constructed to regulate the supply of condensing water corresponding to the temperature and pressure of the compressed gas. The thermostatic control for starting and stopping the motor is so arranged that the motor will stop and start when the temperature in the cooled rooms reaches the two predetermined limits. Another automatic controlling device is a high-pressure safety cutoff and a circuit-breaker.

The subject is worthy of attention and the new business ought to be encouraged by a low rate. A flat meter rate is good, but the machines show up well for both the consumer and the central station if on a sliding scale, depending on the number of hours' use of the maximum demand. The aggregate profit will be large, though made up of small items, when a few of these plants are on central-station lines.

It is the intention of the Massachusetts Institute of Technology to add mechanical refrigeration to its curriculum.

Fundamentals of Refrigeration*

Cold is a relative term, and the degree is measured by the thermometer. The temperature at which water freezes and ice melts is 32 deg. F. To melt ice it is necessary to apply heat, although in the process of melting the thermometer shows no rise in temperature until after the ice is converted into water. The heat necessary to produce this conversion from a solid to a liquid state is called the latent heat of fusion, and is expressed in heat units. The heat unit—that is, the energy necessary to raise the temperature of 1 lb. of water 1 deg. from 62 deg. to 63 deg. F.—is called a British thermal unit, and is abbreviated B.t.u. The latent heat of 1 lb. of ice is about 142 B.t.u. A ton of ice contains therefore

$$142 \times 2000 = 284{,}000 \ B.t.u.$$

This figure has been adopted as the standard of a ton of refrigeration per day, equivalent to the amount of heat necessary to change the ice from a solid to a liquid state. Machines rated at one-ton capacity are able to extract 284,000 B.t.u. during a run of 24 hours, or 11,833 B.t.u. for each hour of operation, this heat-extracting capacity increasing in direct proportion with the increase in capacity of the machine.

Two systems are in use for artificial refrigeration, the direct-expansion system and the brine-circulation system. In the former, the refrigerant piping is exposed within the space to be cooled; in the brine-circulating system the brine is forced through the pipes and returned to the point where the refrigerant extracts the heat from the brine. In the latter case, it is necessary to use a pump to circulate the brine.

To determine the refrigerating capacity of the machine required in any particular case, it is necessary to consider the size of the box, the character of insulation in the walls and the amount of material put into and taken out of the box in a given period of time. A simple and fairly accurate calculation can be made by comparison with the ice consumption.

Suppose a ton of ice is required three times in two weeks during the warmest weather. This is equivalent to 6000 lb. in 14 days or about 450 lb. per day, including Sundays. Assuming that the conditions justify eight hours' daily operation, a one-ton machine operating eight hours daily, or one-third of a day, would give refrigerating results equivalent to one-third of a ton, or 666 lb. of ice. A one-ton machine, although slightly in excess of the requirements, would be the proper size to give results equivalent to the present ice use. Very frequently, however, a lower temperature is desirable, and a larger machine should therefore be recommended.

In the case of a new installation or where no direct comparison with ice can be made, other calculations must be used. This is roughly done as follows: Assume that a butcher with a box 10x10x10 ft., handling 500 lb. of meat per day, delivered to him at about 90 deg. F., desires the box maintained at 34 deg. F. The box is new and the insulation fairly good, the latter having a valuation constant equal to 3. Use this insulation formula:

$$H = cA\ (t - t')$$

where

H = Total heat loss, B.t.u., in 24 hr. = ?;
c = Insulation constant = 3;
t = Temperature outside = 85 deg. (summer heat);
t' = Desired temperature inside = 34 deg.;
A = Outside area of box = 10 × 10 × 6 sides = 600 sq.ft.

$$H = 3 \times 600 \times (85 - 34) = 91{,}800 \ B.t.u.$$

Use this contents formula:

$$F = ws\ (t - t')$$

where

F = Total heat in B.t.u. = ?;
w = Weight in pounds = 500;
s = Specific heat of contents = 0.8 (for beef);
t = Temperature of material put into box = 90 deg.;
t' = Desired temperature inside = 34 deg.

$$F = 500 \times 0.8 \times (90 - 34) = 22{,}400 \ B.t.u.$$

The total B.t.u. to be removed by the machine is therefore 91,800 for insulation added to 22,400 for contents is equal to 114,200 B.t.u. each 24 hours. If the machine is to perform the work in 10 hours out of each 24, what size of machine will be necessary? In the foregoing it was learned that a one-ton machine will remove 11,833 B.t.u. each hour, or 118,330 in 10 hours. Therefore a one-ton machine will do the work in this case, if operated between 9 and 10 hours daily in the hot weather, and a lesser number of hours, in proportion to the temperature prevailing, at other times of the year.

In the formula given, the value of the insulation c will vary from 1 up, 3 being an average condition for fairly good butcher boxes. Where old and poor boxes are used or many windows open into the refrigerator, a higher value must be used, perhaps about 5. The specific heat s varies with different substances. Milk is 0.9, beef 0.8, vegetables 0.9, fish 0.85, eggs 0.75.

*Excerpt from report of committee on refrigeration, ventilation and laundry machinery, read before the National Electric Light Association, at Seattle, Wash.

With her engines of 70,000 hp., the 900-ft. "Imperator," of the Hamburg-American Line, will have a speed of 22½ knots.

A Question in Operation

If a line pump is shut down for repairs, should the compressor be shut down or kept running?

If the expansion of the ammonia takes place in coils submerged in a brine or a freezing tank, the machine may be operated for a short time without any inconvenience. As the temperature of the brine in the tank will be somewhat lowered the operation of the machine should accordingly not be continued long enough to freeze the brine in contact with the expansion coils.

In the case of double-pipe brine coolers the expansion valve should be closed and the machine be stopped practically as soon as the brine pump is shut down. The precaution of stopping the machine in the latter case is necessary because of the limited brine volume surrounding the expansion pipe; freezing it would effect a complete stoppage in the brine circuit and possibly burst the pipe.

In shutting down the compressor for two or three hours, or, in fact, for a much shorter time, the expansion valves should always be closed. This is because the higher pressure behind the liquid in the receiver would continue to force through into the expansion coils, and eventually flood the coils. This ammonia would probably make trouble when the machine is again started by freezing back to the compressor, which means that on account of the abnormally large quantity of ammonia in the coils, some of the liquid will be entrained or drawn back with the ammonia gas returning to the compressor.

It is well to close the expansion valves a few minutes before the machine is stopped when the plant is to be shut down for any considerable length of time. This will reduce the back pressure in the coils to something below normal, so that the rise in pressure which always takes place because of the evaporation of the liquid remaining in the expansion coils will not be much above the usual working pressure.

The suction valve of the machine need not be closed unless the ammonia pressure in the expansion coils rises sufficiently to cause the stuffing-box to leak when the packing contracts on cooling. If the discharge valves of the compressor are not tight, leakage of high-pressure ammonia gas back into the cylinder may also give rise to the same difficulty, when the valve in the discharge line must be closed as well. It is advisable not to close this valve unless it becomes absolutely necessary because the starting of the machine without opening it would probably result disastrously to the machine. There being no outlet for the compressed gas, the pressure might readily rise sufficiently to blow off the compressor cylinder head.

Electric Refrigeration

The committee on refrigeration, ventilation and laundry machinery of the National Electric Light Association presented at this year's convention a classified list of industries using electric refrigeration. Table 1 shows principally the variation in the particular classes of business and the averages in the industries specified. The average tons of refrigeration are 6¾ with an average horsepower connected of 15½, or an average ratio of 1:2.3. The average kilowatt-hours per horsepower connected equals 1241, and the average kilowatt-hours per ton equals 2935.

TABLE 1. CLASSIFIED LIST OF INDUSTRIES

Class	No. of Installation	Total Hp.	Total Tons	Total Kw.-hr.	Kw.-hr. per Hp.	Kw.-hr. per Ton
Butter	2	17	9	16,013	942	1779
Candy	1	7	4	3,402	486	851
Dairy	11	177	73	230,494	1302	3157
Florist	3	30	13	38,638	1288	2972
Fruit	1	7	4	7,654	1093	1913
Hospital	1	75	20	103,376	1378	5168
Ice cream	7	45	19	83,162	1848	4382
Meats	14	285	137	339,561	1191	2471
Poultry	2	25	11	22,423	897	2038
Soda fountain	1	8	3	13,674	1709	4558
Saloon	1	8	3	10,418	1302	3472

Table 2 shows the capacity in tons of refrigeration with the horsepower of the motor required to operate, as found in practice.

TABLE 2. MOTOR CAPACITY REQUIRED

Capacity, Tons of Refrigeration	Hp.
1	3
2	5
3	7½
4	7½
5	10
6	12
7	15
8	15
9	15
10	20
12	20
15	25
50	50

Refrigeration in South America

"Refrigeration is not only a business issue but it has a great moral influence," said John Barrett, director-general of the Pan-American Union, at the recent convention of the American Association of Refrigeration at Washington. "I believe today that refrigeration means a new life for a large portion of Latin America and the tropical countries instead of endless apathy and lack of development.

"The whole Caribbean Coast, the Gulf of Mexico and the Mosquito coast, which a few years ago were the home of yellow fever and malaria, through the influence of refrigeration are being converted into countries where men, women and children can live in comparative comfort."

"What is a vacuum?" said the examiner.

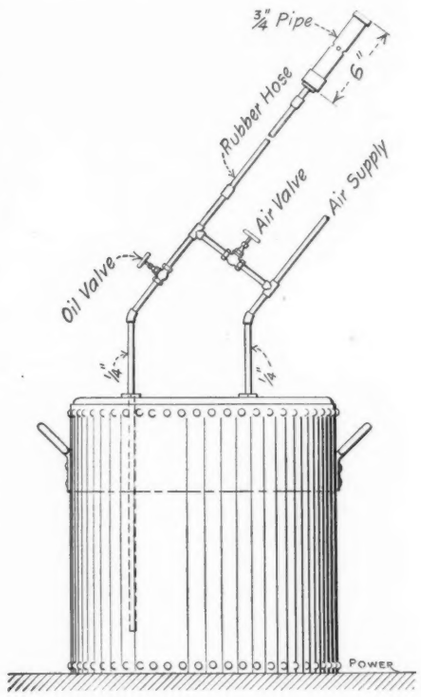
"A vacuum," replied the candidate, as he drew in his breath with a hissing sound, "is a thing that goes like that, and when you have 25 you are all right."

CORRESPONDENCE

Spraying Oil in Absorber Tubes

Every refrigerating engineer has been troubled with leaking absorber tubes while the machine is in operation and in getting the scale out of the tubes when the machine is shut down for winter repairs, but much of this trouble can be avoided by spraying the inside of the tubes with raw linseed oil. They should first be thoroughly cleaned with a flue scraper; then the oil may be applied to good advantage with an oil spray using air. I use the apparatus shown in the sketch. The tank can be placed on top of the absorber and the air piped to it.

This apparatus can be made very cheaply, using any tank that will stand



OIL SPRAYING OUTFIT

a pressure of from 50 to 60 lb. The sprayer is made of ¾-in. pipe for 2-in. tubes and ½-in. pipe for 1½-in. tubes, drilled with four ⅛-in. diametrically opposite holes. To obtain a good spray and not waste the oil, the oil and air valves must be properly adjusted.

After spraying the tubes let water flow slowly into the bottom of the absorber to give the oil on top of the water a chance to cover any unoiled place on the tubes as the water rises.

If this treatment is employed there will be fewer leaks during the season, the scale will not form so thickly on the tubes and they may be more easily cleaned when the machine is shut down for repairs.

GAY A. ROBERTSON.

Louisville, Ky.

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Contents

Page

Limiting Efficiencies of Steam Plants 2
New British Cruiser Makes 31 Knots 3
Substituting Gas and Oil for Coal... 4
Boiler Explosions at Sea..... 4
Combustion and Flue Gas Analysis.. 5
Heating Surface, Boiler Horsepower.. 7
Wrong Gage Connection..... 8
Testing an Engine under Difficulties.. 9
An Unusual Accident..... 10
The Vibragraph..... 11
Vulcan Chain Vise..... 11
Rotary Converters and Their Operation—II 12
Reversed Polarity..... 14
Home-Made Welding Device..... 14
What Made Transformers Heat Up?.. 14
Small Gas Power Plant..... 15
Fuel Supply Troubles..... 16
The Gas Turbine..... 17
Igniter Points..... 17
Automatic Refrigeration an Ideal Central Station Load..... 18
Fundamentals of Refrigeration..... 19
A Question in Operation..... 20
Electric Refrigeration..... 20
Spraying Oil in Absorber Tubes..... 20
Editorials 21-22

Readers with Something to Say: Connecting a Troublesome Muff Coupling....Flue Gases and Combustion....Automatically Emptying a Sump....Improved Steam Piping ....What Ruptured the Boiler?.... Faulty Lubrication....Slip of Eccentric....Securing a Loose Piston.... Accident to Air Compressor.... Reference Blanks.... Home-Made Foot Valve....Steam Trap Experience....Is the Packing Defective?. 23-25

Questions Before the House: Calculation of Heat Value of Coal ....New Valve Effects Economy.... Automatic vs. Fixed Cutoff for Compound Engines....Benefits of Mutual Aid....Instruments and the Engineer....Cotton Rope Drives.... Cable Cuts Shaft....Pump Would Not Force Water....Brown Engine Diagrams 26-28

Inquiries of General Interest..... 29
Study Questions..... 30
Stole Current Worth $100,000..... 30
An Intentional Boiler Explosion..... 31
Asbesteel Gasket..... 32
Regrinding Double Disk Gate Valve.. 32
Ideal Corliss Valve Engine..... 33

POWER

The Growing Use of Superheated Steam

The rapidly increasing use of superheated steam indicates a keener realization of its merits than has hitherto prevailed. The development is at present far more rapid in locomotive than in stationary or marine practice, but the widespread introduction of the turbine is accelerating its use extensively in marine work. Stationary practice cannot long continue so unresponsive to the influence of this development, for generally speaking, what is true of superheated steam in locomotive or marine practice is also true in stationary practice.

In the fifteen years since superheated steam was first used in locomotives in Europe, the practice has grown considerably there, especially in Germany. Happily America is awaking, as one Western road now has over 500 superheating locomotives in service. In marine practice, especially in warships, commendable progress is being made.

Fortunately, the troubles associated with superheated steam are being rapidly overcome by using better metals and design for the superheater and its moving parts in contact with the steam. The great differences in temperature between the outside and inside surfaces of the cylinder or turbine casing containing highly superheated steam, demand uniformity of thickness of the walls, and efficient insulation to avoid excessive contraction and expansion which is apt to stretch the metal beyond its elastic limit.

This particularly applies to all joints and flanges to reduce the likelihood of thermal losses by leakage. Superheated steam is ideal for turbines, as there are no evils incident to lack of thorough lubrication as exists with reciprocating engines, and the normally dry steam acting on the blades obviates the deleterious effects of saturated steam. But, as is pointed out in the next editorial, superheated steam in practice is only *normally* dry and provision should be made to separate the condensation that is bound to form through unavoidable temperature reductions in the steam lines.

The advisability of using superheated steam centers on the facility with which it may be introduced in nonsuperheating plants. The installation cost of the superheater is comparatively low, and there is no mechanism requiring frequent attention and repair. Moreover, the economy gained by reduced water, steam and coal consumption more than compensates for the cost of the installation and maintenance of the superheater.

Superheated Steam

Ever since the importance of superheated steam to economical engine performance was established, most steam users have been content to utilize it without running it through a separator before its admission to the engine. Indeed, most engineers believe it to be unnecessary to do so and are firm in the belief that "Water, no matter how formed, cannot exist in the presence of superheated steam"; that "it robs the steam of its heat and is itself evaporated into steam."

As a matter of fact, wet, saturated and superheated steam may exist simultaneously in the same section of pipe. Time is required to evaporate the water even though the temperature be considerably above that necessary to do so. The existence of ice in an atmosphere above 32 deg. is somewhat analogous.

Modern turbine practice, perhaps more than any other one thing, has shown the ill effects of water in supposedly superheated steam. The blades of turbines normally using superheated steam have been found to be badly eroded after comparatively short service, and it is possible that serious accidents may result from the turbine receiving a slug of water when being started. That the blades were worn considerably in a short time indicates that wet steam was being supplied. The friction set up by the presence of water in the steam also lowers the efficiency of the machine and both effects combine to make frequent and costly repairs necessary. The results of more investigation into the steam conditions of well regulated plants using superheated steam would be interesting.

Small Exhaust Steam Turbines

It is only within the last few years that the exhaust-steam turbine has been utilized in improving the steam consumption of power plants. The installations to which publicity has been given have been of large units and engineers of small power plants may have gained the impression that the adoption of small exhaust-steam turbines is not practicable.

When the load on a small power plant increases beyond the capacity of the engine there are several ways of securing more power. One is to install a larger or additional engine, which would demand additional boiler capacity, and in instances an enlarged boiler house. Another is to install a condenser, if the engine has been operating noncondensing, and obtain an additional output of between 20 and 25 per cent. Installing a low-pressure turbine between the engine and the condenser is the latest way. Small turbines can be operated with a jet condenser, maintaining a vacuum of from 24 to 26 inches, or a surface condenser can be used for higher vacuum. Much of the energy remaining in the steam after passing through the cylinder of a reciprocating engine with its limited range of expansion can be recovered in an exhaust-steam turbine, because the latter greatly extends the range of expansion. Often the turbine gets as much out of the steam as does the engine through which the steam first passes.

Engineers whose plants are being forced should look well into the merits of the exhaust-steam turbine from the standpoints of economy, small space occupied and low initial cost.

Why Isolated Plants Sometimes Lose

Two reputable consulting engineers after investigating the conditions of the plants in the county buildings of an Eastern state, reported that in no plant were records or other data kept from which the cost of power or light might be determined. The plant engineers "could not tell the boiler horsepower developed, the coal consumption per horsepower-hour, the average number of kilowatt-hours developed for lighting or power, the economy of the engines, or the proportion of steam used for heating, lighting and power."

And these plant engineers are civil-service men too. Is it to be wondered that central stations are making inroads into the isolated-plant field? Where central-station service can be supplied for less than the purchaser's own cost for producing power it should prevail, but there are many instances where over-zealous salesmen for that service secure its adoption where it does not mean higher economy, simply because a knowledge of true conditions is lacking.

That the Freeholders did not provide these plants with record forms is no justification of the engineers' ignorance of operating conditions. Had they fortified themselves with the information, the Freeholders would have been humiliated for their own laxity.

It is just as much an engineer's duty to know what his plant is doing as it is for him to keep the knocks from his engines. But so long as such deplorable conditions exist, those at fault must be content to eventually take their place in the "starter and stopper" class.

Precautions for Safety

Constant vigilance is the price of safety, and the plant or shop employee who is ever watchful has small need for liability enactments and workmen's compensation laws. But as familiarity with danger breeds contempt for its presence the employee must be safeguarded; hence it is necessary to compel employers to adopt safety measures and hold them liable for loss of life and limb.

By the installation of safety devices the Pennsylvania Railroad decreased the serious injuries to its 33,242 shop employees over 63 per cent. in ten months. This achievement is due to the systematic methods advised by an insurance company.

Unless plant owners and employees cooperate in reducing the causes of accidents, the best state-made laws will fail of their purpose. Almost any accident-insurance company is equipped to advise an owner how he can reduce his accident hazard, but "the problem of safety," says Vice-president Atterbury, of the Pennsylvania Railroad, "is not altogether a question of rules and safety appliances; it is the development of inherent self-restraint and control. If we can provide instruction in these principles, we will have taken a long step forward in conserving life and limb and increasing industrial safety."

Office Building Service

When the fifth convention of the National Association of Building Owners and Managers convenes in Seattle on July 11, it is to be hoped that in the discussions on construction, maintenance and service, the office-building power plant will have its share of consideration.

Striving after the "superficial attractions" tends to direct the owner's attention to high-speed elevators, expensively equipped toilet rooms, marble-walled corridors, etc., making him forgetful that most tenants are satisfied with prompt responses to elevator calls, sufficient heat and light—in a word, service.

This class of service is supplied by the power plant. The successful manager knows how much is dependent upon his chief engineer, and when the owner reduces his appropriation the manager is too wise to let the economy affect the power plant.

Good power-plant service has often been the means of attracting tenants dissatisfied with buildings having merely superficial attractions, and in keeping the plain office building on a paying basis.

Sunday Work in Power Plants

Now that the summer is here and the beach and field are alluring, Sunday work in hot boiler and engine rooms seems especially disagreeable. But Sunday work must be done. Unfortunately for them, some engineers have to work Sundays to furnish the power that means comfort and enjoyment to the public.

As long as Sunday work is necessary it should be made as easy and agreeable as possible. It is folly to leave until Sunday, work that could be done as well on week days. Those so favored as to have fresh air and sunshine at least once a week, if in charge of men less fortunate, should have consideration for their being confined long hours in underground rooms, especially boiler rooms, and endeavor to ameliorate uncomfortable conditions. The men will not only be pleased with the management but will cheerfully do better work.

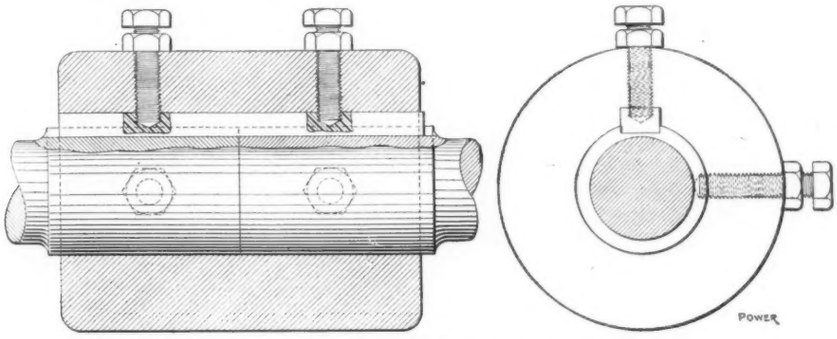
Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Correcting a Troublesome Muff Coupling

A part of the shafting illustrated stopped running, and examination revealed that the key had come out of the coupling, and that both keyways in the shaft and coupling were badly worn.

The shaft that had worked out of the coupling was bent and had moved about in it until ridges nearly $\frac{3}{16}$ in. deep were worn in its surface. New keyways were cut by hand and the keys fitted. On



MUFF COUPLING, SHOWING KEYWAYS AND SETSCREWS

starting the engine the shaft wobbled considerably and after running for three-quarters of an hour the keys were noticed to be gradually coming out. They were then made secure by putting 1-in. steel setscrews, drilled and tapped through the coupling and half way through the keys as shown. Similar setscrews with locknuts were screwed into each section of the shaft at right angles to the keys.

This held good for about two months with an occasional tightening when new shafting was installed.

O. G. A. PETTERSSON.

Manitoba, Can.

Flue Gases and Combustion

It is often stated that insufficient air supply is the cause of CO in the flue gases. Except for a few minutes at a time when the steam pressure is falling, CO is formed by this cause in practical operation.

If CO is present, air enough to supply 24 lb. of oxygen could burn 12 lb. of carbon, one-half to CO_2 and the other half to CO, and generate 113,400 B.t.u. Air enough to supply 21 lb. of oxygen could burn 7.9 lb. of carbon to CO_2 and generate 114,000 B.t.u. Now, by so controlling the fire that CO_2 and not CO is made, 3 lb. less of oxygen is needed to give the same total heat. The 113,400 B.t.u. needed can be obtained with less air if the fire is so controlled as to make CO_2 instead of CO, unless the additional air and the air containing 3 lb. of oxygen does not enter into combustion at all, or, all of the heat generated by the added air and by the 3 lb. of oxygen is added as sensible heat to the loss up the chimney.

I do not believe that with more air supplied less will be used, and all the heat generated by the excess air will not go up the chimney. If these conclusions are true, then steam cannot be kept up with insufficient air, though this is conducive to the most economical fire. If CO is formed the remedy is not more air, but a different control of the fire, perhaps more air above the grates and less below, or a thinner fire. If a remedy is sought by simply supplying more air, more steam will be made instead of less coal burned. Of course, all these conditions are more or less modified in practice, but the margins are wide enough to justify the conclusion that the best practice is to use the least air and that if CO is formed more air than necessary is being used.

HARRY D. EVERETT.

Fort Apache, Ariz.

Automatically Emptying a Sump

In a plant of 5000 kw. capacity all the drainage from the engines, etc., empties into a sump having a capacity of about 400 cu.ft. when full. During the daytime the sump fills in about 25 min., but at night, when part of the plant is idle, it requires about twice as long. It is emptied by an ejector supplied with steam through a ½-in. pipe.

The intake pipe to the centrifugal circulating water pump passing directly alongside the sump suggested using that means of getting the water out. A 1¼-in. hole was drilled and tapped, and a pipe extended a little over half way to the bottom of the sump. A float from an old flush tank was fastened to the stem of a butterfly valve connected at the lower end of the pipe to prevent emptying the sump and breaking the pump's vacuum. The steam ejector with its individual controlling valve and float was set high and left for emergency service. This arrangement has worked very satisfactorily, and empties the sump at little cost.

CHARLES A. RICHARDS.

Mountaintop, Penn.

Improved Steam Piping

Fig. 1 illustrates a horizontal tubular and a vertical boiler in which steam is generated for an automatic engine. The vertical boiler has an angle stop valve *A*, and a vertical pipe connected to the reducing tee *B*. This tee has a drip pipe intended to discharge into both boilers, but cannot because it is below the water line of the higher boiler.

This boiler primes considerably, lower-

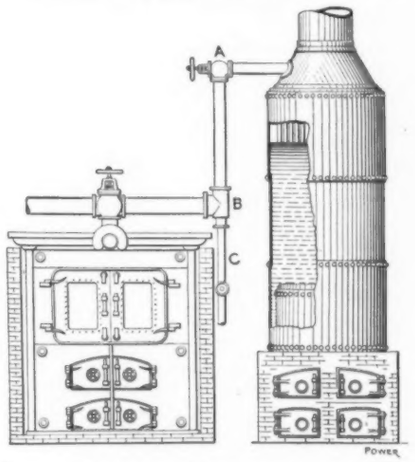


FIG. PIPING OF HORIZONTAL AND VERTICAL BOILERS

ing the water level, and much of the water carried over goes into the lower boiler. If wet steam is supplied by the horizontal boiler, it goes to the engine because it cannot travel against the flow of steam and thus reach the drip pipe *C*.

Fig. 2 shows the changes made in this piping for the vertical boiler. The original stop valve is retained, but a much larger horizontal pipe is substituted above the water line; hence water from priming

and condensation can be returned by gravity to the same boiler. Steam and water flowing through this pipe strike the tee at the left, connected "bull head" for this purpose, and are forced upward, the water being separated from the steam more effectively than with the previous arrangement.

At *A*, Fig. 2, is a continuation of this piping, the ell *B* of which is the same as that shown in the upper left corner. The short horizontal pipe is connected to this ell, followed by another ell *C* and

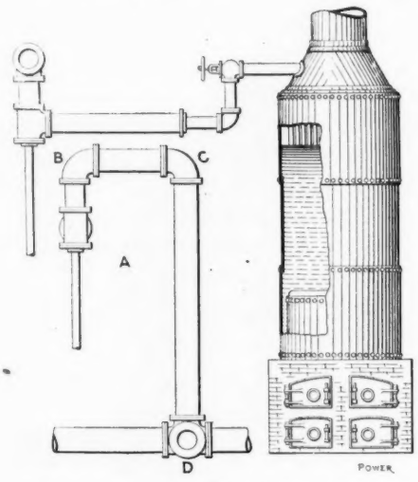


FIG. 2. SHOWING CHANGE OF PIPING

a long vertical pipe which ends in the reducing tee *D;* from this outlet steam goes to the engine. The right opening delivers steam to the heating system, while the left receives steam from the horizontal boiler.

Fig. 3, further illustrating these connections, is a plan of the horizontal boiler showing the angle stop valve *A* connected by a short pipe to the tee *B*. The

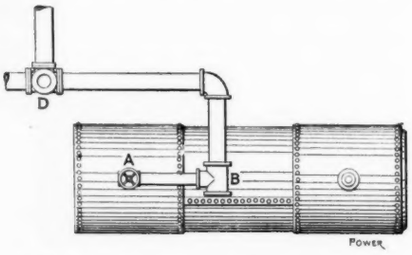


FIG. 3. SHOWING FURTHER CHANGES IN PIPING

horizontal pipe in Fig. 1, originally connected to this tee, was removed and a plug inserted as shown. The reducing tee with the side outlet *D* corresponds to *D* in Fig. 2.

Before these improvements were made, the horizontal boiler and all of the piping shown vibrated excessively because of a slight reduction of pressure in the pipes every time a throttle valve was opened. This loosened much of the brick setting and rendered the whole plant unsafe, but these changes practically eliminated the trouble.

W. H. WAKEMAN.

New Haven, Conn.

What Ruptured the Boiler?

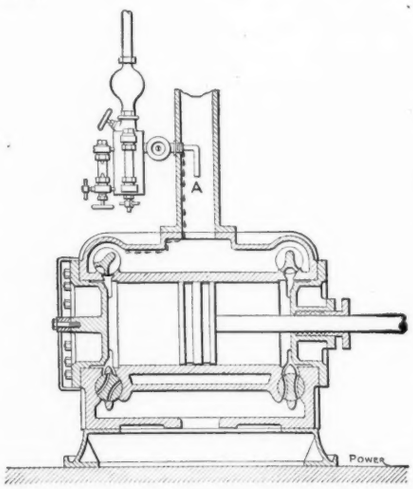
In our plant there are three boilers, each 24 ft. long and working under 80 lb. pressure. The blowoff valve burst on one of them, allowing the water to be blown out. After the boiler cooled off we found a large rupture at the first seam in the firebox and several rivet holes cracked through one to the other for about 3 ft. along the seam. The fire was immediately drawn as soon as we discovered the trouble. Will some interested reader inform me what caused the rupture?

J. W. RIDER.

Rimershing, Penn.

Faulty Lubrication

Unsatisfactory engine performance from faulty lubrication is often due to improperly placed lubricators. The illustration shows a common connection that causes the oil to follow along the side of the pipe and cylinder walls and giving one valve excess lubrication while the others are insufficiently supplied. To correct this a tube carrying the oil should extend to the center of the pipe, as shown at *A*.



LUBRICATOR CONNECTION, SHOWING MOVEMENT OF OIL

One end of the cylinder or valves of a Corliss engine are frequently found worn more than the other, undoubtedly due to improper oil distribution. If the condition obtains it is shown by the engine running poorly.

Some mechanically operated lubricators have no stop valve between the check valve and the cylinder. If the check valve fails to work and lubricator repairs are required, trouble and inconvenience follow from the absence of the stop valve.

C. R. MCGAHEY.

Baltimore, Md.

Slip of Eccentric

These indicator diagrams were taken from the 12x16-in. steam cylinder of an air compressor taking steam at 100 lb. and running at 100 r.p.m. The valve-

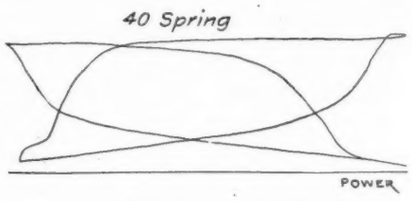


FIG. 1. EFFECT OF ECCENTRIC SLIP

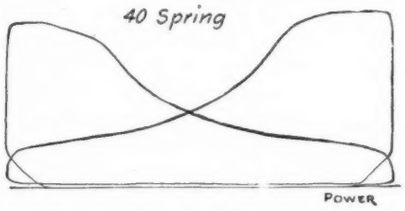


FIG. 2. ECCENTRIC READJUSTED

gear is of the adjustable piston-valve type. Fig. 1 was taken after the eccentric had slipped nearly 90 deg. from normal. Fig. 2 shows the result of properly resetting the eccentric.

WALTER L. HOUSTON.

Republic, Wash.

Securing a Loose Piston

While in charge of the power plant and engineering department of a mine in the mountains of British Columbia, I had an interesting experience on my initial inspection trip with the superintendent. The plant was in a deplorable condition and the superintendent told me to expect almost any kind of trouble.

We were told that the engineer was in the shaft at work on the sinking pump. A few minutes later he appeared, apparently drenched through. He informed the superintendent that the piston was loose again and the pump had stopped. It was a vertical sinking pump of the bucket and plunger type and was hung in the shaft 120 ft. below the surface. This pump drew the water from a sump 10 ft. deep and discharged it to the surface, thus keeping the mine from being flooded. The shaft was sunk about 40 ft. from the bank of the river and the subsoil was sand and gravel.

The leakage in the mine was excessive as the shaft at the time was only timbered. At the bottom the coal vein dipped for 400 yd. at an angle of 45 deg. and consequently if the pump at the bottom of the shaft filled and overflowed, the lower workings would soon be flooded.

It was also found that the steam piston of the pump had previously become loose, due to the stripping of the threads on the rod; attempts had been made to secure it by riveting the rod over the

top of the nut and fitting a taper pin through the nut and rod, but without success. No spare rod or parts were on hand and could not be obtained in less than 10 days, and with this pump out of service the mine would flood in 20 hours. The engineer was instructed to remove the piston and rod and bring them to the surface; meanwhile the writer collected such tools and material as the limited facilities would permit.

The piston was replaced on the tapered rod and a mark scribed on it level with the outer face of the piston, and the projecting end of the rod cut off $\frac{3}{32}$ in. below this mark. This left the small end of the taper 1¾ in. in diameter, into which a ⅞-in. tap hole 2 in. deep was drilled. No stud of suitable size being on hand, one was made from the end of a steel crowbar and screwed in the tapped hole, after which the piston was replaced and a large steel washer ¼ in. thick put on the stud. The piston was forced up hard against the shoulder of the rod, with a nut on the end of the stud to prevent it from backing off. A ⅛-in. hole was drilled in the stud back of the nut and a split-pin inserted. The pump was again in operation three hours after it was dismantled, which is a fair record considering the meager facilities and the difficulties under which the repair was made.

JOHN CREEN.

Seattle, Wash.

Accident to Air Compressor

Recently I was impressed with the fact that considerable care should be taken in removing the head from the air cylinder of an air compressor.

New piston rings were to be put in the piston of the air cylinder. The compressor was not shut down until everything was ready for doing the job quickly, when the discharge valve was closed and the nuts removed from the cylinder head. A chisel was used to force the head from the cylinder and a lighted candle placed on the engine bed to see to work. As soon as the head was loose from the cylinder, gases within, caused by lubricating oil, ignited, blowing the head from the studs and breaking two adjacent pipes. Luckily these were only a 1-in. water pipe and a 1½-in. steam pipe under very low pressure.

J. G. KOPPEL.

Montreal, Canada.

Reference Blanks

While many engineers visit power plants through interest or curiosity, without intending to gather data, they often wish later that they had some particular information relative to these plants. A visitor usually remembers a few facts, such as types, horsepowers or engines, steam-pressure and vacuum-gage readings.

An engineer could profitably equip himself with a few blank sheets arranged in a form similar to that indicated here-

Boilers.....
Steam pressure.....
Feed pumps
Type.....
Size.....
Capacity.....
Water pressure.....
Injection water
Source.....
Discharge.....
Temp. injection.....
Temp. discharge.....
Prime movers
Type.....
Size.....
Hp. or kw.....
Condensing pump
Size.....
Type.....
Dry condenser
Size.....
Type.....
Gage readings
Steam.....
Receiver.....
Step bearing.....
Vacuum.....

with, and jot down what he can remember when he leaves. He will feel amply repaid for his trouble some later time when this information may help solve an engineering problem in his plant.

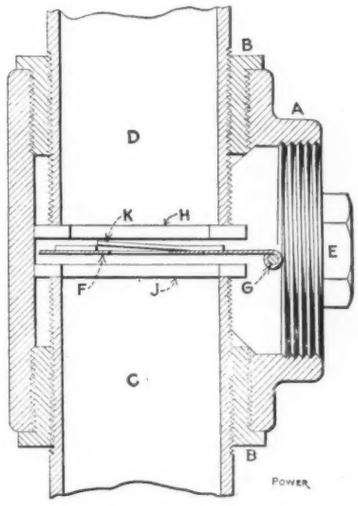
DAVID FLIEGELMAN.

Holyoke, Mass.

Home Made Foot Valve

The foot valve illustrated is one I made for emergency use.

A is a 2-in. tee into the ends of which



DETAILS OF FOOT VALVE

are fitted the 2-in. to 1¼-in. bushings *B*. Into the bushings are screwed the nipples *C* and *D*, 1¼x4 in. and 1¼x6 in. respectively. *E* is a 2-in. plug. The valve *F* is fitted into the tee and swings on the pin *G*. The washer *H* covers the nipple *D* ⅛ in. on its inside edge.

The valve *F* is practically the same as that found in an ordinary hand pump, containing a flapper which is shown at *K*. The action of the valve is obvious.

A. ENGELHARDT.

Newburgh, N. Y.

Steam Trap Experience

We have six bucket-type traps which take care of the condensation in a 14-in. loop and the receiver separators on the engines. The traps discharge into a 4-in. line to a receiver pump, which forces the water back to the boiler. The condensation from the engine jackets runs to a tail trap on the first gallery of the engines and then to a 4-in. drain line in the basement. The tail trap has a vent so the water will drain to the basement. The pressure carried on the low-pressure jacket is 2 lb. The needle valves on the bucket traps gave trouble, due to cutting and grooving, thus putting a pressure on the drain line and forcing the water back up the jacket pipe and out the vent.

Putting check valves on the vent pipes prevented the water from overflowing, but still was not satisfactory. The traps were on a bracket between the steam main and the drain line, so we lowered them 2 ft. below the drain line, thinking that by making the traps discharge against a head the valves would last longer, as they would not hold tight more than two weeks. This new position was very convenient for repairs, but did not help the valves much.

A repairman suggested using a metal other than bronze for the valves, and tool steel was tried. The first or experimental valve is still doing duty and giving excellent results after six months' service. All our traps are now equipped with steel valves.

HENRY R. BLESSING.

Philadelphia, Penn.

Is the Packing Defective?

We are having trouble with the metallic sectional ring packing in the piston-rod stuffing-box of our compound engine. Our efforts to prevent leakage here avail nothing.

The packing consists of six rings, each divided into three pieces, forming scarf joints. A spring fits over each ring which causes it to bear on the rod. A case covering two rings forms part of the packing, and in this instance three cases, each containing two rings, are used. This packing has been in service a little over a year and leaks considerably.

Can some reader suggest a remedy for the trouble?

JAMES PIERPONT.

Oakville, Vt.

An accumulation of one-eighth inch of scale will make a difference of 15 to 18 per cent. in evaporation.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Calculation of Heat Value of Coal

On page 558 of the Apr. 16 issue, Mr. Kowalke gave a method of calculating the heat value of coal from the proximate analysis. The following modification of his formula enables it to be used without the aid of an auxiliary curve.

Mr. Kowalke's formula is:

$$\textit{B.t.u. per pound of coal as received} = 14.760\ C + aV \qquad (1)$$

Where C is the decimal percentage of fixed carbon in the coal as received, V is the decimal percentage of volatile matter in the coal as received and A is a constant which varies for different coals.

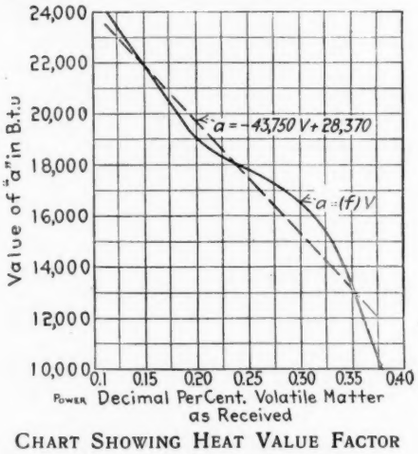


CHART SHOWING HEAT VALUE FACTOR FOR VOLATILE CONTENTS OF COAL

An auxiliary curve giving values for a is necessary for substitution in the formula.

It appears that a is a function of V, but apparently the relation of a to V cannot be expressed mathematically. If it could, there would be no need for the curve. From an observation of the aV function it is seen that a straight line can be superimposed, so that it approximates the relation. The exact location of the straight line is a matter of individual judgment. But having once decided upon it, and assumed that it represents the relation of a to V, it can be stated in mathematical terms, thus eliminating the curve.

I have done this, and found the relation between the variables a and V (as shown by the straight line) to be

$$a = -43{,}750\ V + 28{,}370 \qquad (2)$$

Substituting (2) in (1)

$$\textit{B.t.u. per pound} = 14{,}760\ C + 28{,}370\ V - 43{,}750\ V^2$$

which is the desired formula, in which may be substituted the values of C and V as found from the proximate analysis, without reference to any curve.

It must be noted that this formula gives results which differ somewhat from those found by means of Mr. Kowalke's curve and formula. The maximum difference comes from

$V = 30$ *per cent. to* $V = 33$ *per cent.* where the curve "humps up," and beyond $V = 37$ per cent. The difference at these points may reach 5 per cent.

Solving the example, given in the issue mentioned, of coal containing 40.63 per cent. of fixed carbon, and 34 per cent. of volatile matter

$$\textit{B.t.u. per pound} = 14{,}760 \times 0.4063 + 28{,}370 \times 0.34 - 43{,}750 \times 0.34^2 = 10{,}585.5$$

Mr. Kowalke's formula and curve gave 10,773 B.t.u. per pound. The difference is 10,773 — 10,585.5 = 188 B.t.u. per pound, or about 1.7 per cent.

G. H. CRAWFORD, JR.

Washington, D. C.

New Valve Effects Economy

In the May 14 issue, J. C. Hawkins told of putting in a new valve and securing a quiet running engine with less steam consumption, but he did not state whether he figured out how much outside lap was needed to cause the valve to cut off at half stroke, and if so how, or just guessed it correctly.

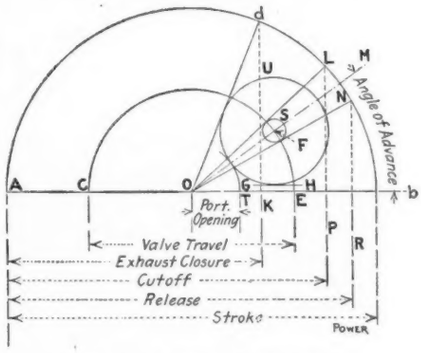
The Bilgram valve diagram would have been very useful to Mr. Hawkins. In a case of this kind the outside and inside lap are known, and if the lead is not known it may be assumed.

To any convenient scale draw on the line AB with O as a center the semicircle ADB having a radius equal to that of the crank (half of the stroke). About O as a center and with a radius equal to one-half the valve travel, describe the semicircle COE. Draw a line GH parallel to AB and at a distance from it equal to the lead. With a radius equal to the outside lap of the valve and with a center F on the semicircle COE describe a circle U that it tangent to GH. About F as a center and with a radius equal to the inside lap of the valve describe a circle S.

Then draw the lines OD, OL, OM, etc., and from their points of intersection with the semicircle, as DLN, etc., drop vertical lines as DK, LP and NR.

With O as a center describe an arc tangent to the outside lap circle and intersecting AB at T. The rest of the diagram is self-explanatory. The angle OMB is the angle of advance. If there is no inside lap no inside lap can be drawn, but perpendiculars dropped from M will indicate exhaust closure and release.

To cause the valve to cut off at half stroke, the lead remaining as before, it is evident that the line OD will be vertical. Describe a circle tangent to OD and GH with its center on the semicircle COE. Now to know the extra outside lap needed to cut off at half stroke, subtract the radius of the first lap circle from the radius of the second circle drawn. This will be the outside lap necessary to add to the old valve to cause it to cut off at half stroke (to the same



BILGRAM VALVE DIAGRAM

scale, of course), or, the radius of the new lap circle, will represent the outside lap for the new valve to cut off at half stroke.

The effects on the other events of the stroke by increasing the outside lap, the lead remaining as before, will cause the center F to be moved farther to the left, and by examination it is seen that exhaust closure will be earlier, release will be earlier and the angle of advance will be increased and, unless the travel of the valve is increased, the port opening will be decreased, because the circle U will be larger and come nearer to O; therefore the distance OF will be less.

A careful study of this diagram will be found very useful. For instance, if a person wished to increase the lead, it is easily seen that all other events of the stroke will take place earlier and to do so the angle of advance must be increased.

B. COESRIB.

Detroit, Mich.

Automatic vs. Fixed Cutoff for Compound Engines

C. J. Mason's letter in the May 14 issue does not give the real reasons for using a fixed cutoff on the low-pressure cylinder of a compound engine. It is because this cutoff is invariable and the reciprocal of the ratio of the two cylinders.

By drawing theoretical diagrams for different loads it will be found that the correct point of cutoff for the low-pressure cylinder remains the same for any high-pressure cylinder terminal pressure. At the end of the stroke in the high-pressure cylinder is a unit volume of steam at the terminal pressure. As shown in Fig. 1, this can be expressed by the product of the volume and pressure. If we call the high-pressure cylinder volume 1, then the energy in the steam doing useful work is $1 \times 30 = 30$. Assuming that the steam follows the simple and convenient law, $pv = constant$, this 30 will be the constant.

Now the steam exhausts and mingles with the steam in the receiver, which is assumed to be large enough to equalize all irregularities. The receiver pressure will then be constant and the same amount will remain in the receiver all the time, the steam doing useful work simply passing through the receiver from the high-pressure to the low-pressure cylinder. The effect of the receiver can therefore be neglected. Then the same amount of steam exhausted into the receiver, $pv = 30$, must be drawn from it to keep its pressure uniform. But the cylinder which draws the steam from the receiver is four times as large; hence the steam available will only fill ¼ of its volume and we have to cut off at ¼. If cutoff is earlier, more steam is put into the receiver than is taken out. This raises the receiver pressure until the equilibrium is again restored, as shown in Fig. 2, where the receiver pressure is higher than the terminal pressure, resulting in a loop.

If cutoff is later, more is withdrawn from the receiver than was put in, resulting in a drop in the receiver pressure, as shown in Fig. 3.

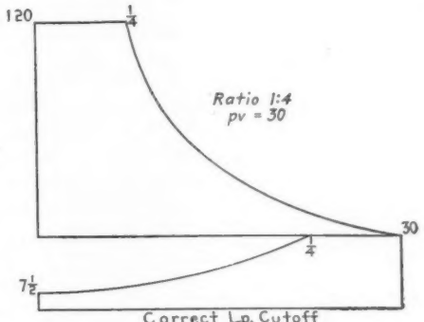
The low-pressure cylinder terminal pressure is fixed by the high-pressure cylinder terminal pressure; it is

$$\frac{30}{4} = 7\tfrac{1}{2}\ lb.$$

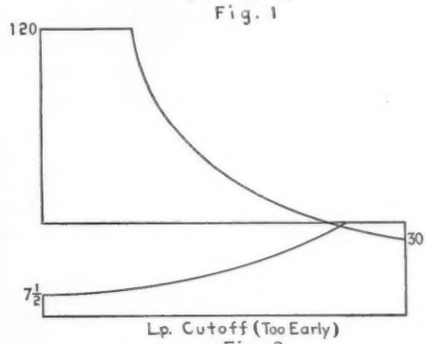
irrespective of the point of cutoff in the low-pressure cylinder. This is in accordance with the above mentioned law. At the end of the low-pressure stroke we exhaust four high-pressure volumes; hence the pressure must be equal to

$$\frac{30}{4} = 7\tfrac{1}{2}\ lb.$$

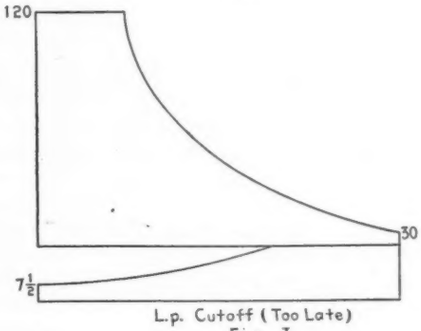
Fig. 4 shows the diagram for a high-pressure cylinder cutoff at one-half stroke. There the amount of steam going through the engine is $\frac{1}{2} \times 120 = 60$ lb.; the high-pressure cylinder terminal pressure is $60 \div 1 = 60$ lb.; the low-pressure cylinder terminal pressure $60 \div 4 = 15$ lb.; the low-pressure cylinder cutoff $15 \div 60 =$ one-quarter stroke.



Correct Lp. Cutoff (Theoretical)
Fig. 1



Lp. Cutoff (Too Early)
Fig. 2



L.p. Cutoff (Too Late)
Fig. 3

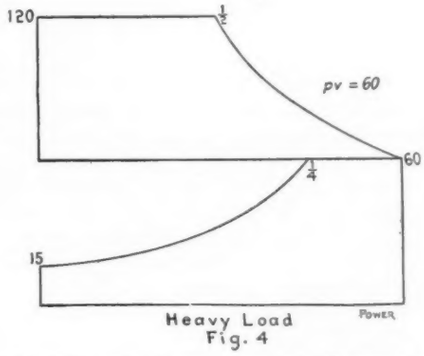


Fig. 4

COMBINATION HIGH- AND LOW-PRESSURE CYLINDER DIAGRAMS

The diagrams are purely theoretical, neglecting clearance, back pressure, etc., and also the fact that in practice it is desirable to have a slight drop at the end of the high-pressure stroke.

F. F. NICKEL.

East Orange, N. J.

Benefits of Mutual Aid

The comment by A. Rauch in the May 28 issue on the Foreword in the Apr. 2 issue is a step in the right direction. If most chief engineers were of the same mind there would be more engineers of higher capabilities. But the man that has to find out things for himself, and has the will to do so, gets there in the end. I have seen the time when I would have given a great deal to have had the chief explain a few things to me, but as he would not I had to find out the best way I could; it took me longer, but I learned it thoroughly. Those calculating to be successful in engineering should spare no reasonable expense or time to develop themselves.

There is a great difference in men. I worked three years in one plant, and had eighteen different fellow engineers with me during that time. I took two engineering journals at that time, and I do not remember of but three men out of the eighteen that took any interest in them aside from looking at the pictures. Each of those has a good position at present; the others are probably just "staying" somewhere.

I know of one instance where a chief engineer lost his position by not helping his assistant. He would not cooperate with his help, refusing to show them or answer questions. The assistant was studious and became a better engineer than his chief, and eventually got the place. If the chief tries to show his help, and answers the questions they ask, they will respect him, and try to live up to the standard he sets, which will more than repay him for the time spent.

V. C. WOOD.

Copenhagen, N. Y.

Instruments and the Engineer

Mr. Martin should receive different advice from that given in John Strong's recent letter. The following incident well illustrates the value of instruments to an engineer.

The plant was under the direct supervision of a master mechanic with two watch engineers working twelve hours each at $2.70 per day. Neither engineer was in charge.

The company was prosperous and able to purchase anything they needed, but could not be interested in an indicator. One of the engineers, more for his own education than the firm's benefit, bought one on monthly payments that he saw advertised in POWER, but with a family of four to support on $2.70 per day it meant a great sacrifice for him. The in-

dicator was put to practical use in setting valves, and bettering conditions in the plant. Later the fuel consumption increased unusually when No. 2 compressor was run; the indicator showed that the high-pressure cylinder was doing no work, as the piston was broken in very small pieces and lying in the bottom of the cylinder; the low-pressure cylinder was receiving the high-pressure steam and doing all the work.

When the master mechanic examined the trouble he noticed the indicator and inquired about it. He found he had a progressive engineer with more intelligence than he had realized. The engineer was made chief with a substantial increase in wages.

This plant has since grown to large size and the engineer who considered the indicator indispensable to his education is a salaried man with one of the best jobs in the city, gets a ten-day vacation every summer, and is in line for higher promotion. Most assuredly it pays to show your employers you are interested in their business welfare.

I have employed many engineers but find most of them as adverse to the use of indicators as they are to mathematics.

W. D. BORRIE.

Richmond, Va.

Cotton Rope Drives

The editorial on the above subject in the Apr. 9 issue calls to mind my experience with cotton ropes when I once tried cotton rope on the strength of what I had heard of its lasting qualities. I installed it on an engine drive, the driving-wheel of which was 22 ft. in diameter, driven wheel 6 ft. diameter, with 48 ft. center to center of the wheels; the rope speed was 5000 ft. per minute and thirty 1½-in. ropes were used.

The grooves were standard, with inclined bottom sides at 45 deg., with curved sides joining the angle. The drive was installed by a cotton-rope expert sent out from the rope mill.

The drive started out in fine shape and the white ropes gliding by made a splendid appearance, but six hours after it was necessary to stop the engine and take up four of the ropes on account of the stretch; this delayed us two hours. We started up again and five hours later we stopped to again take up the ropes.

This continued for a whole week, starting and stopping the plant every six hours to take up ropes, until the entire 30 ropes had all been taken up. A high-priced lubricant was used on the ropes as recommended by the rope expert; he was game, however, and stayed with us.

The second week started in with more delays and more taking up of ropes. One of the ropes broke during the third week and we replaced it with a manila rope. During the past six months that these cotton ropes have been in use, as fast as they give out we have been replacing them with manila ropes, the use of which has never given trouble. From my experience a manila rope is by far the superior of the cotton rope.

A. RAUCH.

Pittsburg, Penn.

Cable Cuts Shaft

John T. Chambers' letter in the Apr. 2 issue brings to my mind a similar case of shaft cutting in which a ¾-in. wire cable was used to drive a machine which was about 200 ft. from the line shaft.

An electric motor was installed to run the machine and the cable discarded; the cable was just lifted off from the grooved pulley and allowed to hang on the shaft. No attention was given it for a couple of hours and then a man went to remove it. He found that during the time the line shaft had been running, the cable, which was stationary, of course, had cut a groove in the line shaft over half way through, and, strange to relate, the cable was not injured or worn in the least. The shaft was not renewed as it is still strong enough to carry the load.

JOHN THORN.

London Junction, Ont.

Pump Would Not Force Water

J. W. Dickson in the Apr. 23 issue asks why raising the suction pipe of his centrifugal pump made it work where it would not work before.

Although he says $100 was spent in trying to prime the pump, he does not say whether the suction pipe was clear at the bottom. I believe his trouble was due to the end of the suction pipe being buried in the sand or mud in the bottom of the well, and raising the pipe 4 in. lifted the strainer out of the mud so that it could get water.

About three years ago I had a similar trouble with a sump pump. The motor was direct-connected to the pump in the bottom of the pit, and the pump had a lift of about 5 ft. and a discharge head of about 30 ft. It supplied water to the skid pipes in the furnaces and the waste from these pipes and the rolling mill ran into this sump and overflowed to the sewer. The sump was of concrete and there was a distance of over 2 ft. from the end of the suction pipe to the bottom of the sump to allow for the collection of sediment.

The pump worked fine for a few days, but the water supply began to decrease and finally stopped entirely. The pump was repacked and primed, but it did no good. Examination of the suction pipe showed that the strainer was buried in sand and scale. After the sump was cleaned out, no further trouble was experienced.

J. C. HAWKINS.

Hyattsville, Md.

When Mr. Dickson raised the suction pipe a pocket was formed which held the horizontal length and part of the riser length full of water; in fact, the water in the riser will be at a higher level than that in the raised end of the horizontal length on account of a partial vacuum being formed between the check valve on the discharge side of the pump and the water level in the riser. This makes a short lift when starting.

In such an installation a foot valve placed in the suction line would prevent the formation of a pocket in which sediment collects, thereby reducing the area of the suction pipe. A pocket also makes it difficult to drain the pipe in winter.

HENRY C. BRIGGS.

Jersey City, N. J.

Brown Engine Diagrams

In the June 4 issue, Mr. Chase asks advice about his Brown engine diagram. It would seem at first that the valves were improperly set, but I think if more care had been taken in tracing the cards, things would not look so bad. The crank-end card shows that the steam cam is too far advanced, causing too much lead, which makes the steam line incline downward to the point of cutoff. I would also advance the exhaust cam to give an earlier compression to increase the economy. The head-end card is not so bad as it looks to be. It shows that it was traced over three times, and at no time with the same governor position, and that the paper on the indicator drum had moved upward, causing the card to be taken lower than the crank end. The steam cam on the head end should be advanced because the round corner at the top of the card shows the cylinder did not receive full pressure at the start.

The exhaust cam should also be advanced for more compression to equal that of the crank end. The cutoff will need adjusting, but the crank-end knock-off cam should not be disturbed because in most old Brown engines it is permanently fixed, and the adjusting is done by the head-end taper pin. The head-end cutoff should be made to equal the cutoff on the crank end. If there is no three-way cock on the indicator piping one should not depend much on the cards being equal, because the time taken to manipulate two globe valves is such that the governor will change its position, and one might think that the cutoff was not correct even when it was. After making the above changes, raise the governor balls as far as they will go, and then let them down ½ in. Put a stick between the balls to hold them in this position. Now turn the engine over and see that the valves do not pick up enough to uncover the lap on the valves. They should not or the engine would race if the main belt broke.

A. C. WALDRON.

Revere, Mass.

Inquiries of General Interest

All Questions Must be Accompanied by Name and Address—Not for Publication

Varying Speed of Engine

Can a 14x30-in. Corliss engine, which makes 100 r.p.m., be converted into a good variable-speed engine by installing a variable-speed belt drive for the governor?

H. F. O.

From the difficulty of obtaining a positive speed from almost any kind of belted variable-speed mechanism, the ordinary Corliss-engine governor will not work satisfactorily with any such form of drive. A better way to vary speed within moderate limits, would be to vary the governor loading by changing the position the governor balance weight. Before placing any dependence on speeding up of engine, a test should be made to determine the highest speed at which the cutoff will work properly.

Size of Boiler Steam Pipe

How is the proper size of main steam connection and main steam pipe for a boiler found?

J. McG.

The sizes of main steam connection and steam pipe are determined according to the amount of reduction from boiler pressure which is permissible in delivering the required flow of steam to the point where the steam is used.

In ordinary power-plant practice, more than 5 lb. reduction of pressure is undesirable; and with the usual number of fittings and length of steam pipe employed, greater reduction of pressure is likely to result unless the pipe is made large enough for the steam velocity in the pipe to be as low as 6000 ft. per minute, or 100 ft. per second.

The rule ordinarily employed for finding the area of steam-pipe connection for a power boiler is: multiply the area of the engine piston by the piston speed and divide by the allowed mean steam velocity in the pipe.

Point for Cutting Off

How can the point of cutoff be found which, with a given initial pressure, will produce a given mean effective pressure?

P. E. M.

Assuming that the absolute pressure varies inversely as the volume, which is approximately true with unjacketed cylinders using dry saturated steam, the mean forward pressure per pound of initial will be given by

$$P_m = (1 + \log_e R)\ (f + c) - c$$

where

P_m = Mean forward pressure per pound of initial pressure (absolute);

$\log_e R$ = Hyperbolic logarithm of the ratio of expansion;

R = Ratio of expansion $= \frac{1 + c}{f + c}$;

f = Fraction of stroke completed at cutoff;

c = Clearance in percentage of piston displacement.

Assume an engine with 5 per cent. clearance, cutting off at quarter stroke, 120 lb. absolute initial pressure, exhaust pressure 15 lb. absolute.

The ratio of expansion would be

$$\frac{1 + c}{f + c} = \frac{1.05}{0.25 + 0.05} = 3.5$$

The hyperbolic logarithm of 3.5 is 1.2528. Then the mean forward pressure with an initial pressure of 120 lb. absolute would be

$$120 \times [(1 + 1.2528)\ (0.25 + 0.05) - 0.05] = 120 \times [(2.2528 \times 0.30) - 0.05] = 75.1008\ lb.$$

say 75 lb. Subtracting from this the back pressure, 15 lb., gives

$$75 - 15 = 60\ lb.$$

as the ideal m.e.p. This would have to be multiplied by a diagram factor to allow for falling steam line, rounding corners, compression, etc., the percentage of the ideal diagram which the actual diagram is likely to cover.

This formula cannot be worked backward to find the point of cutoff directly, and the only way is to try successive points until the right one is found. A table, such as will be found on page 115 of Low's "Steam Engine Indicator," enables one to locate it closely.

Size of Steam Exhaust Connections

A 44x80-in. engine is run at 75 r.p.m. What actual diameter of steam and exhaust-pipe connections should be used?

J. D.

The area of the 44-in. diameter piston is 1520.53 sq.in. and the piston speed is $\frac{80}{12} \times 75 \times 2 = 1000$ ft. per minute.

Assuming that an engine of this size would cut off at one-half stroke, or earlier, it would be safe to assume an average velocity for steam in the pipe at the engine as high as 9000 ft. per minute; hence the size of steam pipe at the engine should be

$$\frac{1520.53 \times 1000}{9000} = 168.95\ sq.in.$$

in area or 14.6 in. in diameter. Not to creat an undue amount of back pressure the size of the exhaust pipe should be proportioned for a steam velocity of 6000 ft. per minute; hence the exhaust pipe at the engine should have an area of

$$\frac{1520.53 \times 1000}{6000} = 253.42\ sq.in.$$

or a diameter of 17.96 in.

The nearest larger standard sizes are a 15-in. diameter steam pipe and an 18-in. diameter exhaust pipe.

Safety of Flywheel Rim

If the centrifugal force which tends to disrupt a flywheel rim is directly proportional to its weight, why is it that increasing the sectional area of the rim would not increase the danger of rupture?

C. H.

Because the cross-sectional area, and consequently the strength of the rim, would be increased in direct proportion with the increase of weight.

Atmospheric Pump

What is an atmospheric pump?

W. H. K.

An atmospheric pump is one in which water is raised by the pressure of the atmosphere. The term, atmospheric pump, though rarely used in present-day practice, is a more scientific name for "suction pump."

Weight of Boiler Plates

What is the need of limiting the variation in weight of a unit area of boiler plates? What is the objection to having one long longitudinal seam the entire length of a boiler?

H. C.

Uniformity in weight of a boiler plate per unit of area, is one of the best checks on its uniformity in thickness. A longitudinal seam the entire length of a boiler is objectionable because it results in greater inequality of stiffness and expansion between the opposite sides of a boiler shell than when the length is broken into several girth courses with the horizontal seams staggered.

Study Questions

This Week's Questions
Last Week's Answers

36. What horsepower is required to drive a 9x13-in. ammonia compressor working against 180 lb. condenser pressure and 15 lb. suction pressure at a speed of 90 r.p.m., assuming a volumetric efficiency of 65 per cent.?

37. A single-acting pump has leaky discharge valves so that for each forward stroke when it discharges 3 gal. it draws back 2 gal. on the return stroke. It makes a forward stroke in ⅗ of a second and a return stroke in ⅖ sec. Starting at the end of a return stroke with water at the bottom of a standpipe holding just 22 gal., how long will it be before water is forced to the top?

38. A building one mile from the source of electrical supply has two thousand 60-watt lamps and a motor load of 50 hp. The voltage at the building is 550 and the drop is 5 per cent. of the initial voltage. What is the ohmic resistance per mile of the line from the power house?

39. A fireman had been having trouble with the injector on one of his boilers for a number of days but finally succeeded in fixing it. When asked how long it was out of order and how long since it was fixed he replied: "When it is the day after tomorrow I fixed it a week before yesterday and my trouble with it began the day after five weeks before a fortnight and three days after the day before a couple of days after I fixed it." How long had it been repaired and how long was it out of order?

40. What is the tensile stress per square inch in a 24-in. pipe, ⅜ in. thick, under a pressure of 80 lb. gage?

Answers to the above will appear in the next issue. Answers to last week's questions follow:

31. Let x = the total number of lamps and a, b, c and d the number of lamps in each row when there are three, four, five and six rows respectively.

Then

$x - 1 = 3a$ $\qquad x = 3a + 1$
$x - 2 = 4b$ $\qquad x = 4b + 2$
$x - 3 = 5c$ $\qquad x = 5c + 3$
$x - 4 = 6d$ $\qquad x = 6d + 4$

A direct solution cannot be obtained as there are only four equations and five unknown quantities, so the solution must be obtained by trial. It is best to begin with d, as that must be the smallest of the unknown quantities, and assume different values for it beginning with 1, until a value for x is obtained which satisfies the other equations, remembering that all of the unknown quantities are whole numbers and x is less than 100. When $d = 4$, $x = 28$ and the third equation is satisfied, but neither of the other two. The next value for d that will bring c out a whole number is 9, x then being 58, and this also satisfies the first two equations; hence is the right answer. The only number of rows into which the lamps can be divided evenly is two.

32. Imagine the stairway unwound and stretched out as a straight line. It becomes the hypothenuse (h) of a right triangle having an altitude (a) of 200 ft. and a base (b) of 5 times the circumference of the tower.

From

$$h^2 = a^2 + b^2$$
$$(282.8333)^2 = (200)^2 + b^2$$
$$b^2 = 39{,}994$$
$$b = 199.98 \text{ ft.}$$

The circumference of the tower is

$$\frac{b}{5} = 39.996 \text{ ft.}$$

and the diameter

$$\frac{39.996}{\pi} = \frac{39.996}{3.1416} = 12.731 \text{ ft.}$$
$$= 12 \text{ ft. } 8\tfrac{3}{4} \text{ in. nearly}$$

33. Absolute zero measures an imaginary condition where no heat is present. It has been determined by calculation to be —460 deg. F. or —273 deg. C. Therefore, 60, 100 and 247 deg. F. would be 520, 560 and 707 deg., respectively, above absolute zero, and 0 and 100 deg. C. would be 273 and 373 deg. above.

34. At 62 deg. F., 2 gal. of water weighs

$$8.3356 \times 2 = 16.6712 \text{ lb.}$$

Then

$16.6712 \text{ lb.} \times 9 \text{ deg.} = 150.0408$ B.t.u.

is the amount of heat lost by the iron to the water and corresponds to a reduction in temperature of

$$\frac{150.0408}{0.13} = 1154.16 \text{ deg. F.}$$

The original temperature of the iron therefore must have been

$$71 + 1154.16 = 1225.16 \text{ deg. F.}$$

35. Using the prismoidal formula

$$V = \frac{h}{6}(A_1 + 4Am + A_2)$$

where

V = Volume;
h = Height;

A_1, Am and A_2 = Top, middle and bottom areas, respectively;

the volume of the chimney, if solid, would be

$$V_1 = \frac{32}{6}\left[\frac{(40)^2 + 4\left(\frac{40+56}{2}\right)^2 + (56)^2}{144}\right]$$
$$= 516.74 \text{ cu.ft.}$$

The volume of the space inside the stack is

$$V_2 = \frac{32}{6}\left[\frac{(24)^2 + 4\left(\frac{24+40}{2}\right)^2 + (40)^2}{144}\right]$$
$$= 232.29 \text{ cu.ft.}$$

The actual volume therefore is

$V_1 - V_2 = 516.74 - 232.29 = 284.45$ cu.ft.

and the weight, at 120 lb. per cubic foot, is

$$284.45 \times 120 = 34{,}134 \text{ lb.}$$

Stole Current Worth $100,000?

Two men were arrested in New York City on June 21, charged with defrauding the Brooklyn and New York Edison companies, causing a loss of $100,000.

According to the press reports, the men, an ex-policeman and an Edison electrician, had the assistance of dishonest users of electricity. Much of their recent work has been done on Coney Island, where they are said to have been in collusion with 50 users of electricity.

The officials are not sure whether these dishonest users will be proceeded against, but they threatened to turn off the lights in the 50 places where they know they were cheated. A police inspector says that one of the accused men would go to a dishonest consumer and offer to cut the bills down. When the financial agreement had been reached he would take a key that opened meters, and with a certain piece of mechanism, turn the meter back. He would visit a place just before the company's reader came around, and the amount lost would be almost the whole sum due. A $500 bill has been reduced to $100. It was the inspector who estimated the losses of the companies at $100,000.

The Edison companies have known they were being defrauded for the last 11 months, and four months ago they asked the detective bureau to find the men responsible. The detectives who made the arrest, it is said, saw the men tampering with the meter of a big Coney Island dance hall.

Similar contrivances have been used before. In 1907 the police arrested four persons who had tried them. They were simple, but required an expert knowledge of meters to use them.

A man arrested in 1907 showed at his trial just how his contrivance was used. He had discovered that by disconnecting a switch and attaching a "ground wire" the meters would not register most of the electricity that passed through. With this device a customer might use $20 worth of electricity in a month and the meter would show only $2 worth. The man pleaded guilty and was sent to jail for a year.

An Intentional Boiler Explosion

A demonstration, which promises to revolutionize the construction of locomotive boilers, was made at Coatesville, Penn., on June 20, under the direction of Dr. W. F. M. Goss, Dean of the College of Engineering, University of Illinois.

FIG. 1. "BOMB-PROOF" FROM WHICH BOILERS WERE OPERATED DURING TESTS

Two full size locomotive boilers, designed for high-speed heavy passenger service, were each subjected to severe low-water tests. Both boilers were identical in size and design, except that one had a Jacobs-Shupert sectional firebox, built by the Jacobs-Shupert U. S. Firebox Co., while the other had an ordinary radial stay firebox.

For the purposes of these tests, both boilers were mounted in a field and 50 ft. apart, and were operated from a "bomb proof" located 200 ft. away from the nearest boiler. Oil was used for fuel because of the danger to which a fireman shoveling coal would have been exposed. Fig. 1 shows the "bomb proof" and on top may be seen the telescope for reading the graduated water gages and pressure gages on the boilers, as shown in Fig. 2.

In the process of testing, each boiler was brought to a condition of operation closely approaching maximum power estimated to be 1400 hp., which is equivalent to that required to haul a heavy passenger train at 60 miles per hour. The supply of feed water was then shut off, but all other conditions were continued unchanged.

The water level gradually fell under control, exposing the crown-sheet and other portions of the heating surface to the full effect of the fire. The boiler having the Jacobs-Shupert sectional firebox was continuously tested under these severe conditions for 55 min. without developing any failure, notwithstanding the fact that the level of the water fell to a point more than 25 in. below the crown-sheet. The water gage-glass did not read below 25 in. The test was then discontinued because the small amount of water remaining did not evaporate sufficiently fast to supply the draft necessary to maintain the fire. At the conclusion of this test, the Jacobs-Shupert firebox was apparently in good condition, and ready for further service, as may be seen in Fig. 4.

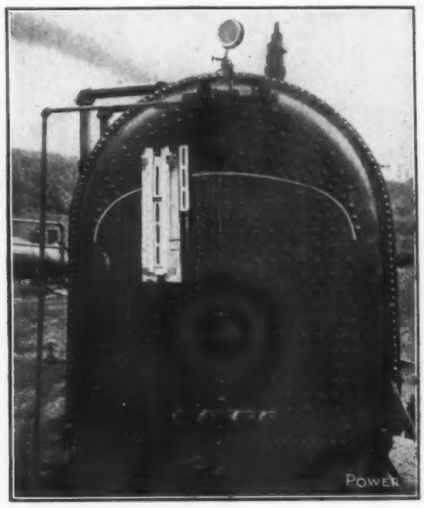


FIG. 2. GRADUATED WATER LEVEL GAGES READ THROUGH TELESCOPES AND PRESSURE

The ordinary radial stay boiler was then tested under conditions identical to those above described. After the test of the ordinary boiler had been in progress for 23 min., and the water level had fallen to 14½ in. below the crown-sheet, an explosion occurred. The crown-sheet and the stays, which hold it in place, having become highly heated, pulled away from each other and released the pressure in the boiler. The discharge of steam was through the firebox and the force of the explosion amidst clouds of steam and smoke, was sufficient to throw parts from the furnace in all directions

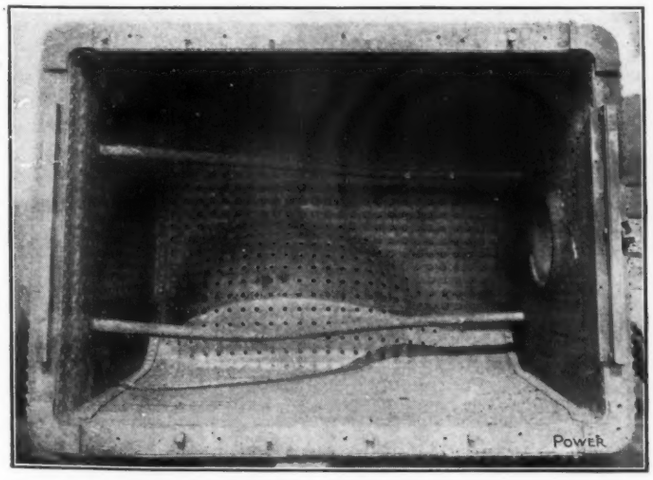


FIG. 3. INTERIOR OF RADIAL STAY FIREBOX AFTER LOW-WATER TEST

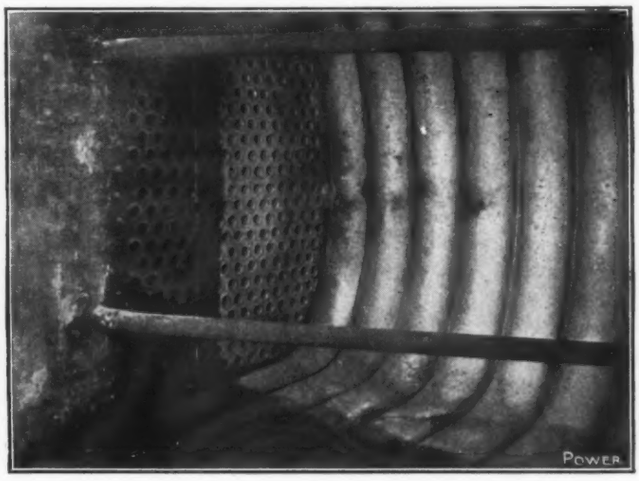


FIG. 4. INTERIOR OF JACOB-SHUPERT FIREBOX AFTER LOW-WATER TEST

for a considerable distance, and to lift the entire boiler, weighing 40 tons, several feet above its foundation. The damage to the boiler was such as to make necessary its reconstruction. Fig. 3 shows the failure of the crown-sheet and Fig. 5 is a view taken when the boiler exploded.

Some of his tests showed an evaporation as high as 50 lb. per square foot of heating surface. The efficiency of the Jacobs-Shupert boiler was a little better than that of the ordinary radial-stayed boiler against which it was tested.

The Lukens Iron & Steel Co., of Coatesville, has financed the Jacobs-Shupert

FIG. 5. EXPLOSION OF THE RADIAL STAY FIREBOX BOILER

The steam pressure during the Jacobs-Shupert test varied between 215 and 225 lb. for the first 27 min. and gradually dropped thereafter until it reached 50 lb. at the end of the test. The steam pressure on the radial boiler varied between 220 and 230 lb., being 228 lb. when failure occurred.

The Jacobs-Shupert firebox is constructed of U-shaped sections between which are riveted stay sheets as shown in Fig. 6. Previous tests by Prof. Goss show that from 45 to 50 per cent. of the evaporation occurs on the firebox surfaces with moderate capacity and some 35 per cent. when forcing.

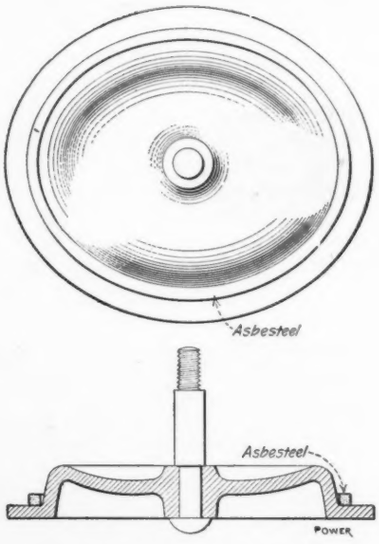
United States Firebox Co. and provided the means for carrying on these expensive and important tests.

Asbesteel Gasket

This gasket is made up of asbestos inclosed in a steel casing and in suitable sizes and shapes to fit the handhole and tube plates of water-tube boilers.

It is suitable to use as a union gasket and also for Van Stone joints. The illustration shows its application to a water-tube boiler tube plate.

As the gasket is compressible it is squeezed flat when brought under pressure between a handhole or tube plate and the boiler shell or header. The steel casing forms a metal-to-metal contact between the header and the tube plate; consequently there is nothing to adhere to either, and when the joint is broken, the gasket falls off, leaving a clean surface. This eliminates scraping



ASBESTEEL GASKET ON TUBE PLATE

and cleaning the plate or header. The gasket is manufactured by Faul & Timmons, 286 Ellicott Sq., Buffalo, N. Y.

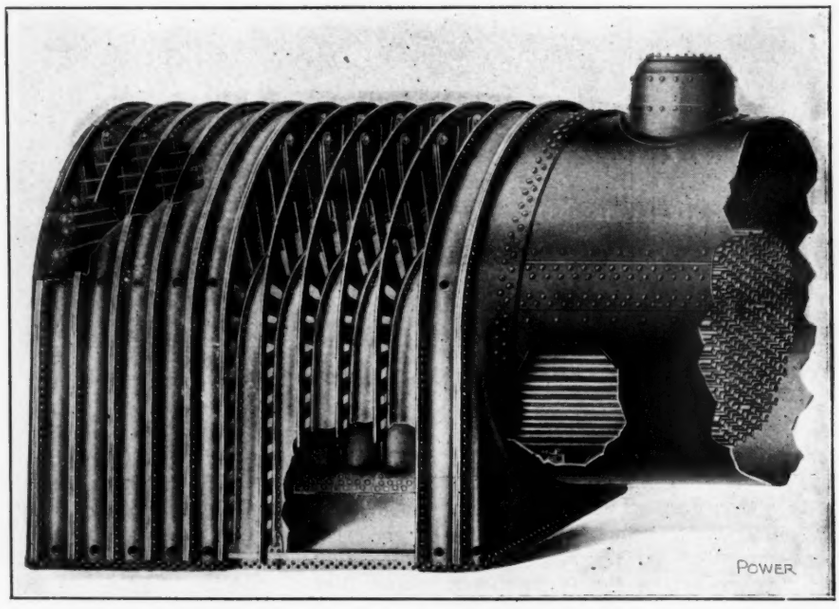


FIG. 6. BROKEN VIEW OF THE JACOBS-SHUPERT FIREBOX SHOWING CONSTRUCTION

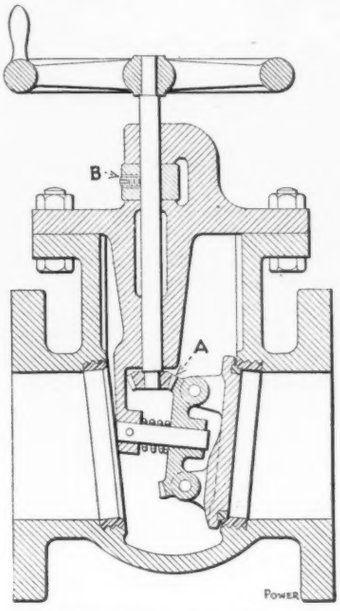
Regrinding Double Disk Gate Valve

This valve is a combination of the globe-valve seat and straightway body which can be reground without removing the valve from the line.

The double-disk feature, although not new, is so constructed that the valve can be used in any position without the faces of the disk and valve body constantly rubbing while being operated.

When regrinding, the bonnet and working parts of the valve are removed from

the body. The valve disks are detached from the head and, one at a time, are attached to the bevel gear *A*. The grinding fixture is attached to the body, as shown by two bolts, allowing the adjusting nut *B* to slip on the shaft. The disk will readily find its seat and the adjusting



REGRINDING DISK VALVE

nut is then tightened to allow the bevel gears *A* to run freely.

As the opening in the upper part of the body is but partially closed by the grinding fixture, the disk can be seen and plenty of room left for inserting the grinding mixture on the seat. This valve is the invention of F. K. North, Pottstown, Penn.

"Ideal" Corliss Valve Engine

One of the first of a new line of Corliss-valve engines built by A. L. Ide & Sons, Springfield, Ill., is shown on the testing floor in Fig. 1. The valves are placed in the cylinder heads, to reduce the port clearances and minimize steam losses.

The valve-gear is nondetachable, and the steam valves are operated by a positive valve motion, Fig. 2, which permits the valves to remain in a closed position for one-half of the revolution, while the eccentric is crossing the dead center and reversing its direction of travel. When the eccentric crosses the other dead center the valves which are balanced are open. The valves are moving at their highest velocity when the eccentric is approximately on the quarter and is imparting the maximum speed to the eccentric rod and connecting gear. They thus call for the greatest driving power when the most is available.

The multiported valve-gear links are inclosed and submerged in an oil-tight case, on the frame of the engine, dispensing with the connecting rocker shafts between the eccentric rods and the valve rods.

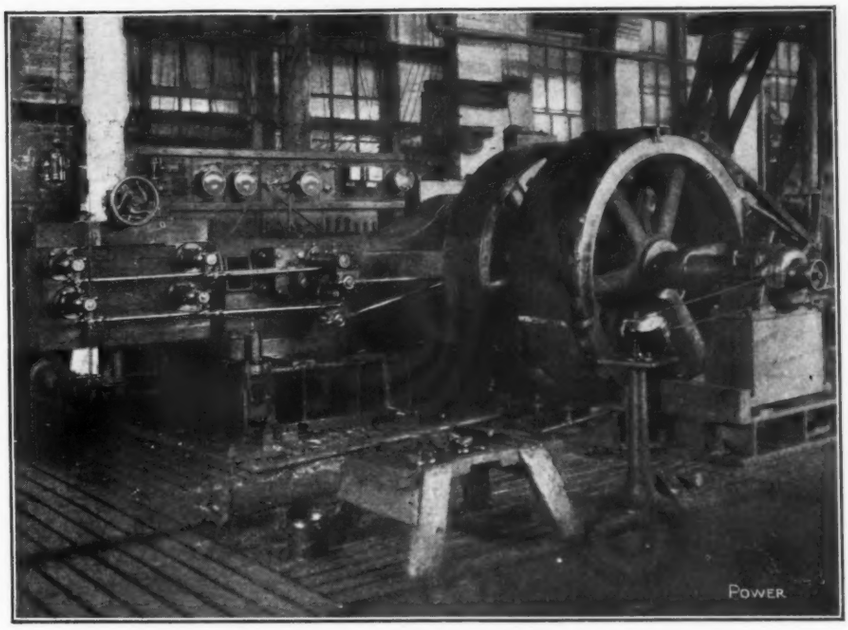


FIG. 1. "IDEAL" CORLISS VALVE ENGINE ON TESTING STAGE

The exhaust valves are driven by the ordinary wristplate motion because they have to be open during nearly one-half revolution and cannot remain motionless as long as the steam valves. The latter are controlled by a shaft governor and the exhaust valves by a fixed eccentric. The steam eccentric operates the lever *A*, Fig. 2, within the case by an external arm, shown in Fig. 1. The lever *A* actuates the pins *B* and *C* through links as shown. The links *D* slide through the guide pins *E* which are fitted in bearings in the valve-gear case and cover. Fig. 6 shows the details of the links *D* and guide pins *E*. The rotation of the pins *B* and *C* is transmitted to the valves through the external lever arms and rods, shown in Fig. 1.

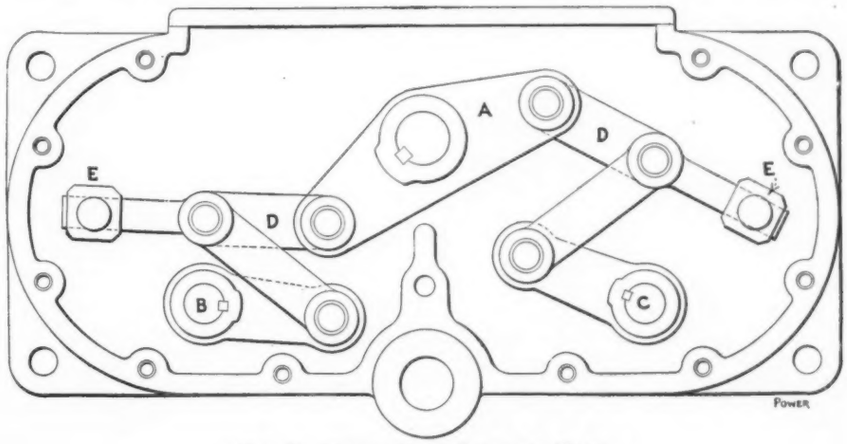


FIG. 2. VALVE GEAR LINK IN CASE

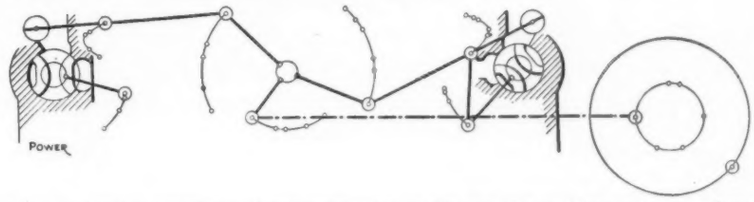


FIG. 3. DIAGRAM SHOWING ARRANGEMENT OF VALVE GEAR USED ON THE STEAM VALVE

In the valve-motion diagrams, Fig. 4, the outer circles at the right represent the crankpin path and the inner, the eccentric travel. In the top diagram the

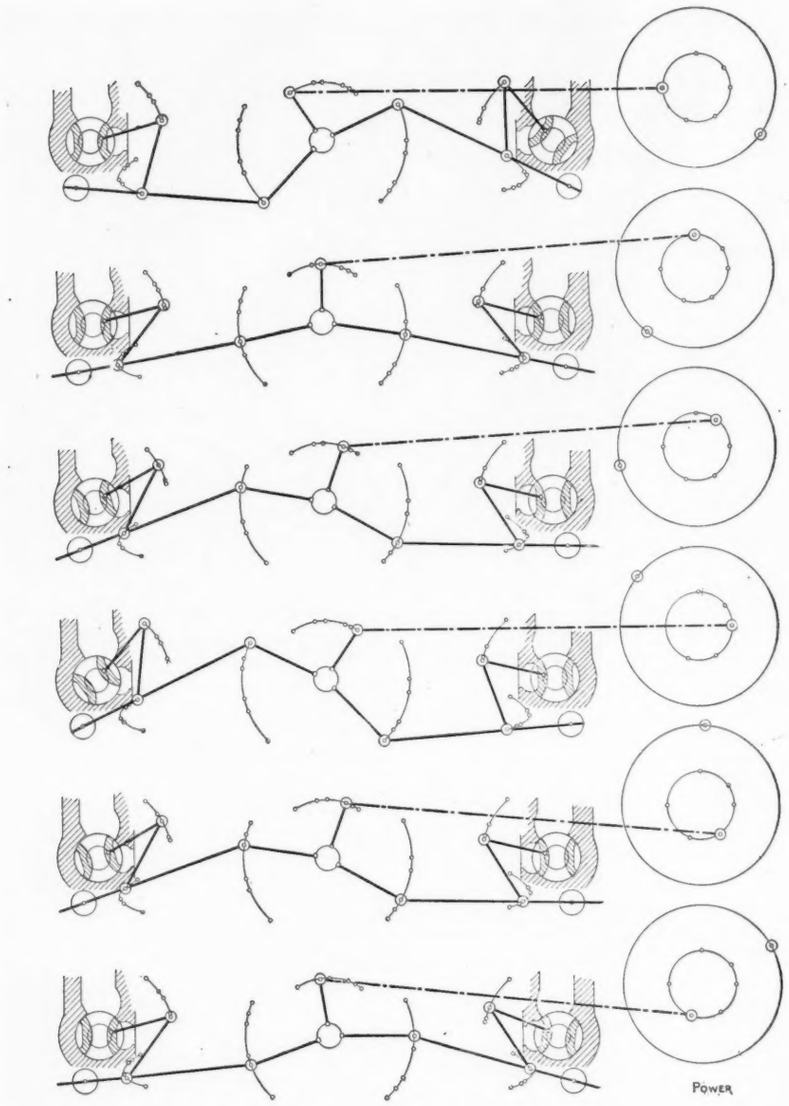


FIG. 4. VALVE GEAR MOTION DIAGRAMS

eccentric is on dead center, the head-end valve is closed and at rest and the crank-end valve is at maximum opening. The position of the valves and the gear at the various other points in the eccentric travel is shown by the other diagrams. The gear in actual use is inverted from that shown, as the gear will work in both positions, as illustrated in Fig. 3.

Typical maximum-load indicator diagrams from the 16x22-in. engine under test, are given in Fig. 7. The steam pressure was 147 lb. gage, speed, 200 r.p.m., and the mean effective pressures, head end, 109.75 lb., crank end, 97.29 lb. per square inch.

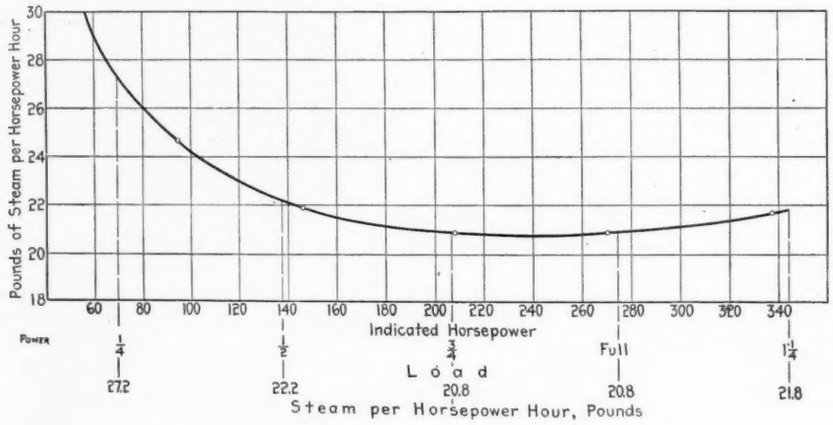


FIG. 5. STEAM CONSUMPTION CURVE OF TEST ENGINE

In Fig. 5 is shown the steam consumption of the test engine at various loads. The pressure was held as close as possible to 150 lb. gage throughout the test, not varying over 4 lb. above or below.

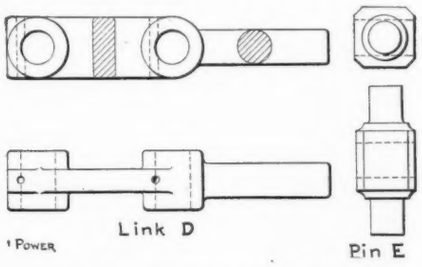


FIG. 6. DETAILS OF LINK AND GUIDE PINS

No correction for moisture was made in determining the steam consumption, as the steam was supplied by a water-tube boiler having no heating surface above the water line, and the superheater was flooded, to prevent superheating the steam. A separator was placed just above the engine throttle. As a further check against the steam carrying superheat a thermometer was inserted in the cylinder steam chest and the temperature and gage readings taken simultaneously every five minutes. In each instance the observed temperature was about 6 deg. below the corresponding temperature of saturated steam of the observed pressure. This difference is chargeable to radiation from the thermometer cup and the exposed length of the thermometer. The indicator diagrams and water records proved that the load was held practically constant, as the planimeter readings did not vary over 2 per cent. in any case and the water records checked equally close.

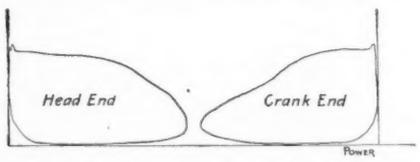


FIG. 7. MAXIMUM LOAD DIAGRAMS

Platt Iron Works Reorganization

In July, 1911, the Platt Iron Works Co., Dayton, Ohio, went into the hands of receivers and its affairs have since been administered by them. Under the plan of reorganization, recently indorsed by a large majority of its creditors, ample working capital for development and improvement will be provided. The necessary amount has been subscribed and is now available.

It is estimated that the net manufacturing profit, from July 24, 1911, to Mar. 1, 1912, without allowing for depreciation above ordinary repairs and maintenance

nance or for interest on investment, was $47,672.96, or about $79,000 per annum. It is believed that upon reorganization and under competent and aggressive management the volume of business can be largely increased and certain economies of operation effected which will materially increase the annual earnings of the property.

The failure was brought about through borrowing large sums, wholly or partially on the faith and credit of the company, no substantial part of which was used for its purpose or benefit. The present organization is believed to be a valuable asset which should be preserved, and in the judgment of the bondholders and creditors, their interests will be promoted through a reorganization rather than by the sale and liquidation of the properties, the dismemberment of the plant and the practical abandonment of organization and good will. The plan is designed to give the new company a working capital of approximately $300,000.

Lima N. A. S. E. Convention

On June 20, Lima Subordinate Association No. 21 of the National Association of Stationary Engineers began its state convention in Lima. It was intended to have a three-day session, but the convention adjourned at 12:15 o'clock on June 21, as many of the delegates were anxious to return to their homes.

The convention was formally opened at 10 o'clock on June 20 in Memorial Hall by Chairman J. F. Reed of the local executive committee. The Rev. G. W. Watson delivered the invocation. Mayor C. N. Shook in welcoming the visitors spoke briefly of the great progress made by the stationary engineers and of the moral influence of the organization.

State President John J. Coughlin responded to the mayor. The association was not, in the usual meaning of the term, a labor organization, said Mr. Coughlin, but it was accomplishing its purpose of raising the stationary engineer to the highest standard, making him more valuable to his employer and promoting the safety of the public.

C. H. Wirmel, chief examiner of steam engineers, who represented Governor Judson Harmon, spoke of the advancement in stationary engineering through the efforts of the N. A. S. E., which had been instrumental in having license and inspection laws enacted. Other states, said Mr. Wirmel, are copying the Ohio laws. Under the influence, education and requirements of the organization, stationary engineering had been raised from a drudgery to a profession.

Prof. D. D. Ewing, of the Ohio Northern University, responded briefly. On behalf of the Lima Progressive Association, E. J. Maire, its president, also welcomed the visitors, expressing the appreciation of the business men of Lima for the honor of entertaining the engineers. Judson Pratt, a delegate, outlined the objects and high attainments of the association. He was formerly an organizer in Ohio and the N. A. S. E. representative when Lima Local No. 21 was first instituted. He referred to the heroism of the "Titanic" engineers. "These men," said Mr. Pratt, "did not seek an excuse to go on board the rescue ship 'Carpathia' —they went down with the 'Titanic' without leaving their posts of duty."

John Lawlor, a pioneer stationary engineer of Lima and one of the oldest members of Lima Local, welcomed the visitors on behalf of the Lima organization. Anthony Dietz, of Cincinnati, gave an original toast to the ladies of the auxiliary.

John Roberts, of Cleveland, a member of the state organization and the father of the stationary engineers' license law, dwelt at some length on the high objects and attainments of the order.

The business of the final session on June 21 was concluded by electing the following officers and selecting Toledo as the convention city for 1913:

T. T. Synett, Dayton, Ohio, president; John E. Radigan, Cleveland, vice-president; W. D. Spangler, Lima, secretary; John Wirmel, Cincinnati, treasurer; Charles Flenner, Hamilton, Ohio, conductor; B. A. Held, Toledo, doorkeeper; T. E. McFadden, Columbus, state deputy.

Exhibitors

White Star Refining Co., Crandall Packing Co., Strong, Carlisle & Hammond Co., Keystone Lubricating Co., Standard Oil Co., Diamond Power Specialty Co., Cling-Surface Co., Westinghouse Electric & Manufacturing Co., Vulcan Soot Cleaner Co., Knopf & Johnson, *National Engineer*, Ohio Grease Lubricant Co., C. E. Squires Co., Fairbanks Co., Quaker City Rubber Co., Automatic Steam Trap Specialty Co., Jenkins Brothers, Hooven-Owens-Rentschler Co., William Powell Co., American Steam Gauge & Valve Manufacturing Co., Lagonda Manufacturing Co., Greene, Tweed & Co., Garlock Packing Co., V. D. Anderson Co., Lunkenheimer Co., Clement Restein Co., Wickes Boiler Co., H. W. Johns-Manville Co., Dearborn Drug & Chemical Works, Hawkeye Boiler Compound Co., Anchor Packing Co., McLeod & Henry Co., Home Rubber Co.

OBITUARY

William Northrop

William Northrop, president of the Virginia Ry. & Power Co., Richmond, Va., died suddenly of heart failure on June 11, 1912, at his home near Forest Hill. Mr. Northrop was born in Pennsylvania 43 years ago. He was educated at Philadelphia and at the University of New York. Shortly after graduating, Mr. Northrop accepted a position with the Philadelphia & Reading R.R.

The Richmond Passenger & Power Co. and the Richmond Traction Co. were consolidated in 1902 and the control passed in 1902 to the Gould interests. Mr. Northrop was assistant secretary-treasurer until 1904, when he and Henry T. Wickham were appointed the company's receivers. On July 1, 1909, the Virginia Ry. & Power Co. succeeded the Richmond Passenger & Power Co., Mr. Northrop being elected president. He was also an officer in several other companies. He is survived by a widow.

SOCIETY NOTES

The headquarters of the American Society of Heating and Ventilating Engineers during the Detroit convention on July 11 and 12 will be at the Hotel Tuller.

The Massachusetts State Convention of the N. A. S. E., which takes place this year at Northampton, July 11 and 12, promises to be larger than any yet held in the state. Earnest work by the various committees has perfected all of the arrangements.

PERSONAL

Frank Salomon, president of the Otto Gas Engine Works, Philadelphia, Penn., sails on July 4 for a six weeks' trip in Germany.

O. D. Dillman, formerly in the main office of the Hooven, Owens, Rentschler Co., Hamilton, Ohio, has been appointed assistant manager of the Chicago branch.

Edgar D. Sibley recently resigned as assistant to the superintendent of power of the Brooklyn Rapid Transit Co. and is now associated with W. G. Clark, consulting engineer, New York City.

D. L. Fagnan, for some time an erecting engineer with the De La Vergne Machine Co., has been assigned to the Cincinnati office of that company in charge of sales for the De La Vergne oil engine.

Paul M. Warburg, of Kuhn, Loeb & Co., New York City, was elected a director of the Westinghouse Electric & Manufacturing Co., at the annual meeting in East Pittsburgh. He succeeds Charles A. Moore, resigned.

S. G. Meek has been appointed assistant general manager of the electrical department of the H. W. Johns-Manville Co., New York City. He has been with the company as special representative in this department over 15 years.

Samuel Insull, president of the Commonwealth Edison Co. of Chicago, recently elected a director of the Pacific Gas & Electric Co., of San Francisco, was given a banquet at the Fairmont Hotel, San Francisco, on the evening of June 14.

Moments with the Ad. Editor

Charles H. Parson, treasurer of the Parson Manufacturing Company, sent us a letter recently which we are reproducing here because it is another definite example of some of the things we have been trying to point out on this page.

Mr. Parson wants a "prize."

Not because he needs it but because he is the kind of a man that when he goes into something in the shape of a contest he likes to win for the sport's sake.

He writes us as follows, with his letter headed—

"Does 'Power' advertising pay?"

"In response to your original suggestion of advertisements that attract attention, we at that time wrote you advising that we thought our advertisements were reasonably strong and attractive and gave our reason. You didn't send us any prize but it may interest you to know that the following are facts:

"In February we received an inquiry regarding the Parson [illegible] Ejector, to which we replied by mail as the point was nearly 2,000 miles distant. On receipt of our letter and booklet we were favored with a night telegram asking price and giving details. On March 14 we closed the order and on March 30 the material was shipped with working blueprints. May 5 the material had arrived and on June 4 in response to our statement, and a request for their advice as to whether they had any difficulties in erection, we received the following reply:

" 'Everything in, working satisfactorily. Will send check to cover in a few days.'

"Your Mr. Robbins was shown this letter which happened to arrive on my desk at the time of his visit, and can bear witness to the authenticity of the statement herein mentioned, although names have been purposely omitted because we have neither asked for nor received the consent of our customer to use the letters above referred.

"On reconsideration, if you find that we have won a prize, send it along."

We didn't offer any prizes, Mr. Parson, but if we had you would have undoubtedly won one with this letter.

We did offer to pay for those letters which were printed and this one will be paid for.

You readers of "Power" may not know it, but one of the hardest things a publisher has to do is to extract definite information from an advertiser as to results received.

Mr. Parson didn't send us this letter with an idea of getting a prize, or being paid for it. He sent it because he was broad enough between the eyes to know that this testimonial to "Power's" pulling power would be appreciated and would be encouraging to the publishers of the paper.

There are two strong points brought out here, one for readers and one for advertisers.

In the first place, a concern 2,000 miles distant from the manufacturer found exactly what they were evidently looking for in the way of an ash ejector. Their night telegram and the rapid closing of the sale is evidence that they were in *need* of such a device. Their need was supplied through "Power" advertising. And they were just as well satisfied to find the advertisement as the concern was to sell the goods.

The goods are pronounced by them "satisfactory." Both parties to the transaction benefited. Way off from this part of the world some subscriber of "Power" has seen the wisdom of reading the advertising pages and has confidence in advertised goods. His confidence was not misplaced.

The other point of view—that of the advertiser—is this:

A large piece of power plant apparatus has been sold through the mails without the aid of personal salesmanship. As a general rule this is a thing which cannot be done. But, in this instance it was successfully accomplished because the live wire who is a subscriber to "Power" knew that he could gamble on anything which appeared in the advertising columns of this paper.

Remember that it pays to read advertising and buy advertised goods, because only reliable goods can be continuously advertised.

An advertising bluff is called quicker than any other.